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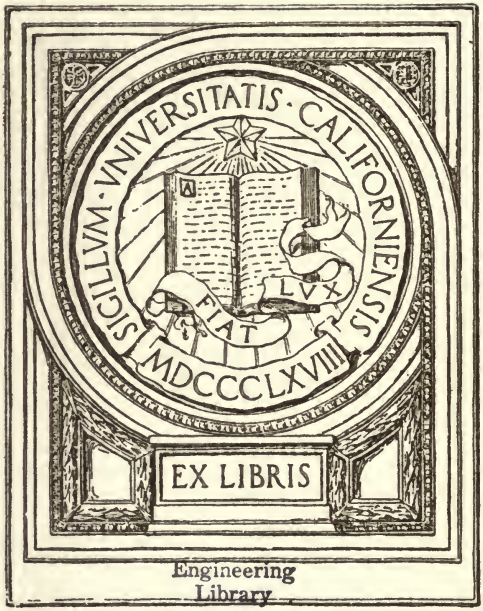

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No. 959.*

A NEW OIL-TESTING MACHINE AND SOME OF ITS RESULTS.†

BY ALBERT KINGSBURY, WORCESTER, MASS.

(Member of the Society.)

1. It has become well recognized by experimenters that there are two important properties of lubricating oils and greases on which their value depends—viscosity and body. Viscosity is the property by virtue of which the lubricants form comparatively thick films between rubbing surfaces, permitting perfect lubrication, a property which is well understood and capable of precise measurement. Thurston early pointed out the case of perfect lubrication as probably capable of exact mathematical treatment,‡ and Reynolds, in 1886, published an elaborate analysis of such a case,§ with application to the experiments of Beauchamp Tower.‖

2. As the result of computations from Tower's data, Reynolds found that the minimum thickness of the oil film and the difference of the radii of the brass and the journal were .000375 and .00077 inch respectively, the load being 100 pounds per square inch and the diameter of the journal 4 inches. In 1899 the writer experimented with a journal 3.82 inches in diameter and 10 inches long, bearings and journal having exactly the same radius. The chords

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* Presented at the New York meeting (December, 1902) of the American Society of Mechanical Engineers, and forming part of Volume XXIV. of the *Transactions*.

† For further references on the same topic see *Transactions* as follows:

No. 7, vol. i., p. 73: "Measurement of the Friction of Lubricating Oils." C. J. H. Woodbury.

No. 120, vol. iv., p. 315: "Economy in Lubrication of Machinery." Geo. N. Comly.

No. 163, vol. vi., p. 136: "Measurements of Friction of Lubricating Oils." C. J. H. Woodbury.

No. 173, vol. vi., p. 437: "Valuation of Lubricants by Consumers." R. H. Thurston.

No. 404, vol. xi., p. 1013: "Durability of Lubricants." J. E. Denton.

No. 433, vol. xii., p. 405: "Special Experiments with Lubricants." J. E. Denton.

‡ "Friction and Lost Work." Wiley & Sons.

§ "Theory of Lubrication." *Phil. Trans.*, 1886.

‖ *Proceedings of the Inst. Mechanical Engineers*, 1884.

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of the bearing surfaces were 3 inches each, the speed 80 and 190 revolutions per minute, and the journal was flooded with machinery oil. By measurement of the displacement of the bearings the oil film was found to have a mean thickness of from .00021 to .00023 inch under loads varying from 27 to 270 pounds per square inch.

3. A condition essential to the formation of such films, as shown by Reynolds, is that the rubbing surfaces should have a very slight inclination to each other in the direction of their relative motion. This condition is generally fulfilled by a slight difference in the radii of the journal and the bearing, due to the original looseness of fitting or to wear. When the loads are very great or the surfaces irregular, or when the conditions are otherwise such as to make the necessary inclination impossible, it is well known that the action of the lubricant is imperfect and frequently very defective. In such cases the theory of Reynolds, which so clearly accords with experiment for the conditions of perfect lubrication, becomes quite inapplicable; nor has any theory been formed which applies to these imperfect or extreme conditions. Such cases occur in pivots, where the surfaces are necessarily parallel, in cylindrical bearings which are too closely fitted, in any portion of any bearing surface where the pressure is unduly high, and in heavily loaded bearings generally. Under any of these circumstances the effect of the lubricant in reducing friction depends mainly upon the "body" or "oiliness." The nature of this property, or combination of properties, is not well understood, but it appears probable that it is an intensified viscosity in that part of the fluid within the region of attraction of the surface molecules of the metal.

4. One of the most frequent causes of contradictory results in friction tests is that the effects of viscosity may readily mask the effects of body, and another cause is found in the changes in the rubbing surfaces which always take place with wear, and which must, therefore, accompany any test for body.

In any well-fitted journal in which perfect lubrication exists, the friction is determined by the speed, the pressure, and the viscosity of the oil. Varying any one of these factors while keeping the others constant, there is some value of the variable for which the coefficient of friction is a minimum * and which at the

* Thurston: "Friction and Lost Work," pp. 311, 326. Woodbury: "Measurement of Friction of Lubricating Oils," *Transactions* A. S. M. E., vol. vi., p. 151.

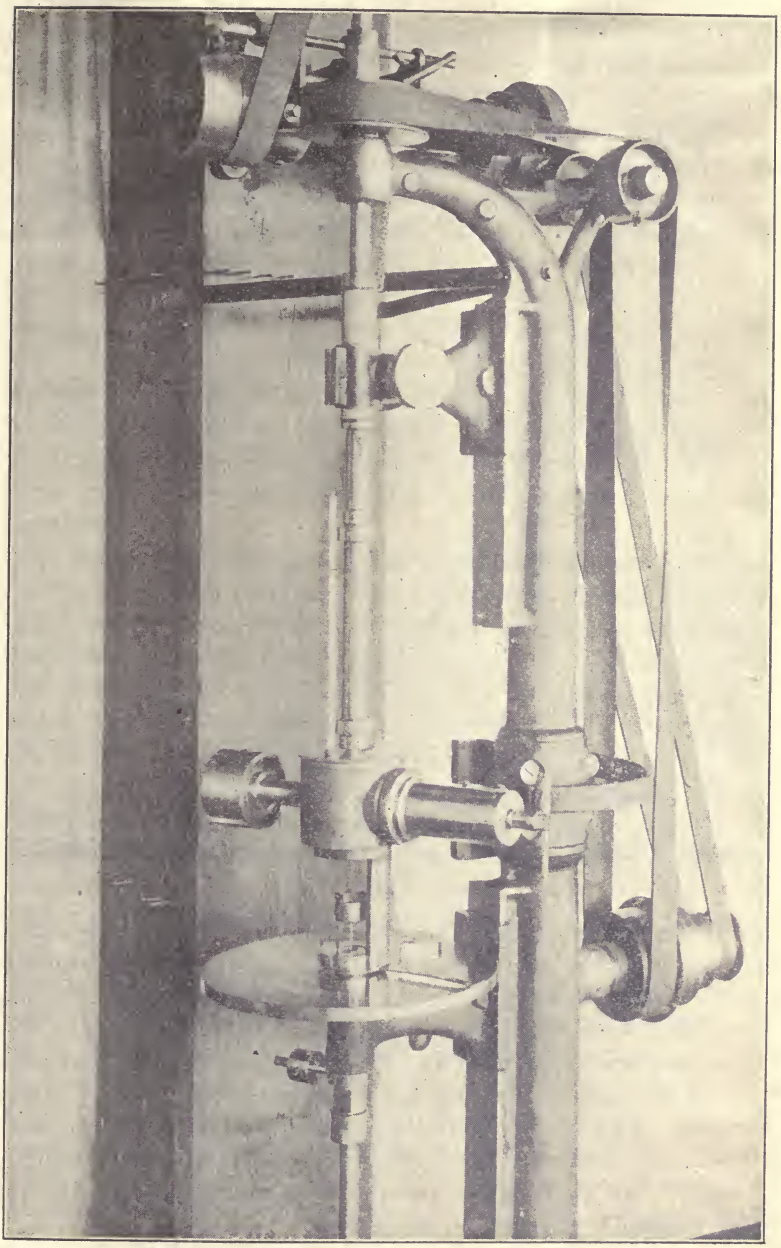

FIG. 33.

same time marks very nearly the limit of the variable for the condition of perfect lubrication. This is illustrated by the curves plotted in Fig. 34, representing tests made on the machine described later. In these tests the load and the speed were kept constant, the viscosity of the oil being varied in each case by varying the temperature. For each of the three oils, the minimum coefficient of friction is reached at a temperature of about 180 degrees Fahr., with the given speed and pressure. As the

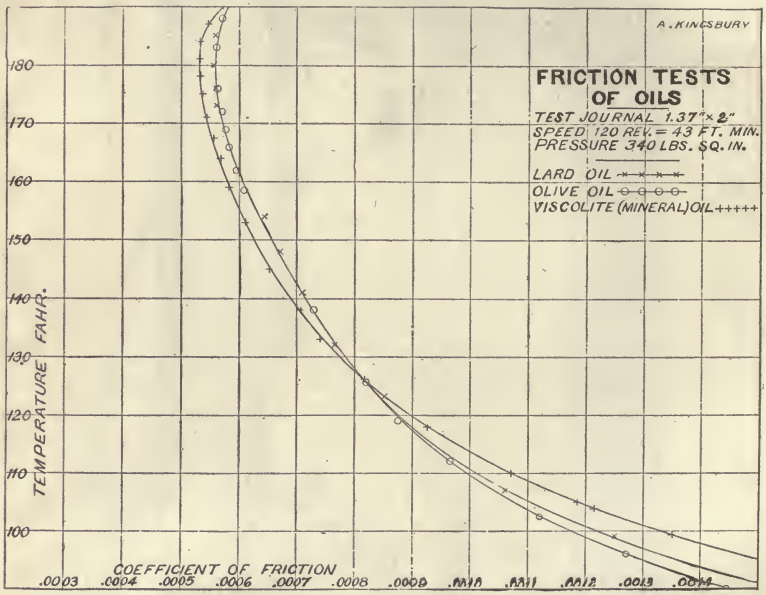

FIG. 34.

temperature varies either way from this value, the coefficient of friction increases; on the one hand increasing with the viscosity of the oil, the metallic surfaces being completely separated by a measurably thick film; on the other hand increasing because the decreased viscosity permits the surfaces to approach so that some parts of the nominal bearing areas are subjected to very intense pressures, up to the limits of strength or plasticity of the metals. It is in these localities that the body of the lubricant determines to some extent the friction and the wear of the journal, the viscosity being also effective on some parts of the area.

5. The relations of the coefficient of friction to the other variables, in a journal giving results as plotted in Fig. 34, may be stated as follows:

With increase of	Where the viscosity is effective, the coefficient of friction	Where the body is effective, the coefficient of friction
Pressure	Decreases.	Increases.
Speed	Increases.	Decreases.
Temperature	Decreases.	Increases.
Viscosity	Increases.	Decreases.
Body	.....	Decreases.

It is thus seen that the effects of body and of viscosity are in nearly all respects diametrically opposite, and that it must necessarily be very difficult to derive reliable information regarding the lubricating values of oils from friction tests in which the effects of viscosity and those of body are not separately recognizable. Under this consideration, methods of testing which are described in this paper are arranged with special reference to the conditions under which the effects of either property may be investigated independently of the effects of the other. The apparatus used is serviceable also for tests under any intermediate condition.

6. Fig. 33 shows the general appearance of the testing machine, for the frame and driving parts of which a 14-inch drilling machine was utilized. The test journal has its axis vertical; it is suspended from the spindle by means of a flexible coupling and runs between two opposed bearings in a cylindrical cup or case, which may be filled with the oil to be tested if a "bath" is desired. The load on the bearings is provided by means of a helical spring of 900 pounds capacity, with screw adjustment and with a device for quick application or removal of the load without disturbing the adjustment. This spring is enclosed in a horizontal tube attached to the side of the oil case. The cup has a cover with a small hole for the insertion of a thermometer.

7. The cup and attached parts are borne on a hollow vertical spindle $1\frac{3}{8}$ inches in diameter, turning freely in a sleeve supported from the frame of the machine; the spindle extends about two feet below the sleeve and is suspended from a fixed bracket by a tempered steel wire passing through the spindle to its lower end. In testing, these suspended parts turn freely to a position where the torsion of the suspension wire balances the friction at the test journal, and the angle of torsion, which may be as great as 270 degrees, is read from a graduated disk. The suspended parts being counterbalanced, there is no appreciable pressure of the spindle against its sleeve; and when the oil in this bearing becomes evenly distributed, there is no error from friction. as has been

amply proven by tests with an "optical lever" as well as by the uniformity of the results in use. At the same time, the viscosity of the oil serves the purpose of damping the oscillations which arise from variations in speed or friction at the test journal. This mode of suspension gives large indications for very small frictions at the test journal, while a helical spring placed on the extension of the spindle is added for tests involving great friction.

8. The cup and the test journal contained in it may be heated as desired by a Bunsen flame. The revolutions of the journal are indicated by a counting device, not shown in the figure.

9. For tests involving perfect lubrication (friction due to viscosity only), the test journal used is $1\frac{3}{8}$ inches diameter, of tool steel, hardened, ground, and polished. The brasses are sectors cut from a ring finished in the lathe, each having an arc of about 120 degrees and a length of 2 inches. These brasses are fitted with some care, so that when perfectly clean they may be made to adhere to the journal after the manner of well-fitted "surface plates." In making tests, care is taken to prevent wear of these parts, which are used only under such loads that the oil film effects complete separation of the surfaces and entirely prevents wear; the load is always relieved before starting or stopping the journal; and, finally, a friction device in the driving coupling safeguards the journal from motion against excessive friction. These precautions against wear are necessary to insure the constancy of results.

10. It will be noted that the above conditions are not such as occur in practice; but it should also be noted that this method of testing is intended to show the effects of viscosity in lubricants and not the varying imperfection of any particular bearing surfaces subjected to wear.

11. In Figs. 34 to 37 are shown curves plotted from tests made with the test journal and bearings just described, a bath of oil being used in each instance. Figs. 34, 36, and 37 show the effects of variation in viscosity upon the friction, both by the use of oils of different viscosities and by the variation in viscosity due to change of temperature. In Fig. 34, the lard oil and the olive oil have very nearly the same viscosity, the lard oil being slightly more viscous, as is also and more clearly shown in Fig. 35; the mineral oil (Fig. 34) is more viscous than either at low temperatures, but less at high temperatures (see also Fig. 38). Fig. 36 shows the friction of four cylinder oils with varying temperatures;

Fig. 37, four engine oils. Fig. 38 shows relations of friction, speed, and viscosity for five oils at three different speeds; the viscosity was found by the ordinary and rather crude pipette method. These curves, although relating to conditions not far removed from the limit of capacity of the journal, approximately verify Reynolds' deduction that the friction is proportional to the viscosity of the oil, and they also show the friction to vary roughly as the square root of the speed. The data for Fig. 38 were taken from average results of tests made in the laboratory of the Worcester Poly-

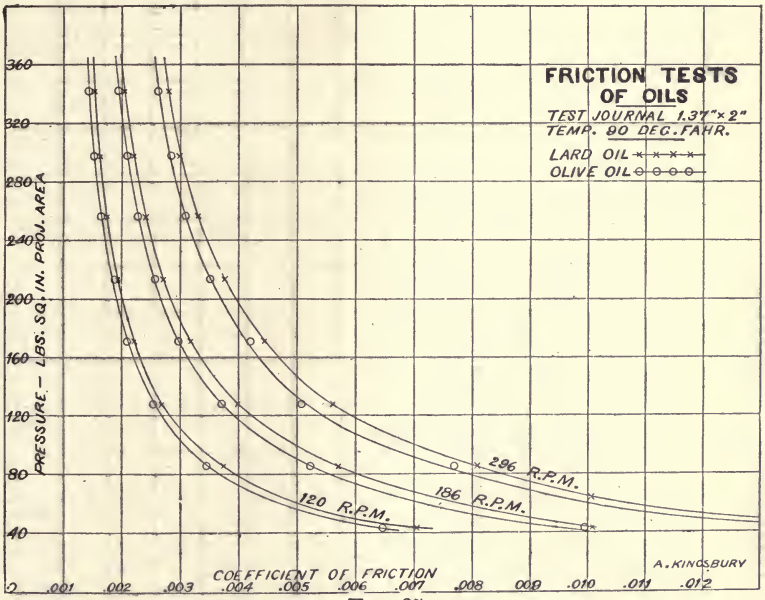

FIG. 35.

technic Institute, by members of the class of 1901; the degree of certainty attained in the friction tests may be illustrated by the following values of the coefficient of friction found by the successive experimenters, no corrections being made for variations in speed nor for errors in calibration of the torsion wire.

COEFFICIENTS OF FRICTION FOR SPERM OIL; PRESSURE, 340 LBS. SQ. IN. 90° FAHR.

117 R. P. M.	182 R. P. M.	281 R. P. M.
.000906	.001171	.00146
.000915	.001158	.00155
.000886	.001158	.00151
.000915	.001125	

It may be noted that the lowest coefficients of friction shown in Figs. 34, 36, and 37 are extremely small, and the writer believes them to be smaller than any hitherto recorded for lubrication maintained solely by the motion of the journal. The minimum friction is but one-fifth of that of the best ball bearing, as far as

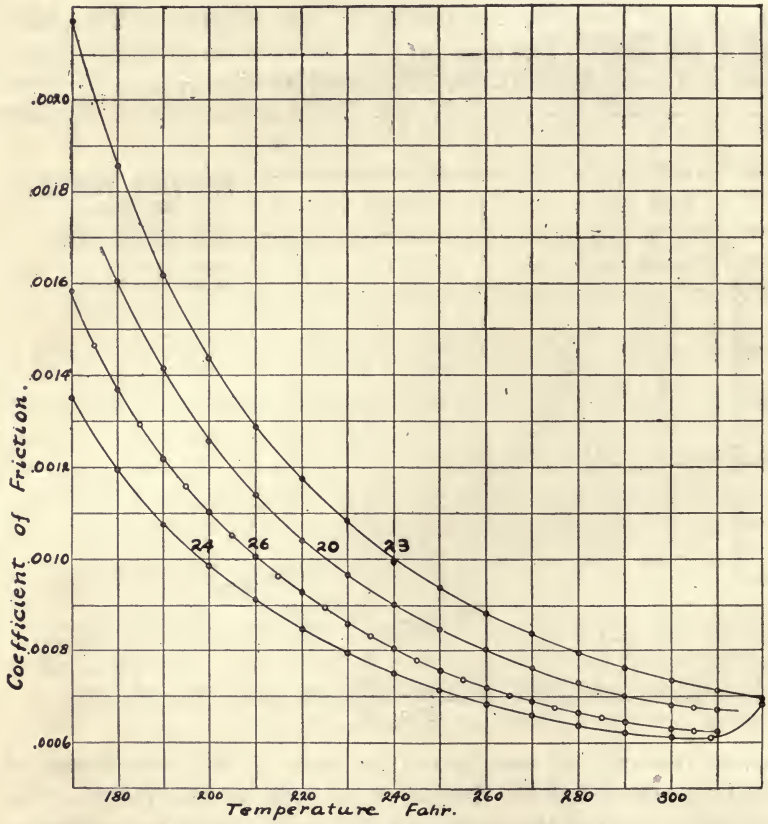

FIG. 36.—FRICTION TESTS OF CYLINDER OILS.
PRESSURE, 340 POUNDS PER SQUARE INCH. SPEED, 94 FEET PER MINUTE.

tests of the latter have been recorded. The value for the minimum coefficient for all oils tested on this journal has been found to be approximately .0006, whatever the speed, pressure, and temperature by which the minimum coefficient may be determined; the oils varying from "spindle" to "cylinder," the speeds from 42 to 101 feet per minute, the pressures up to 340 pounds per square inch and the temperatures up to 340 degrees Fahr. Very

nearly the same minimum coefficient was found in the writer's tests * of a journal lubricated by air only; the value in which case was .00075.

12. For tests for comparing oils with respect to body or oiliness,

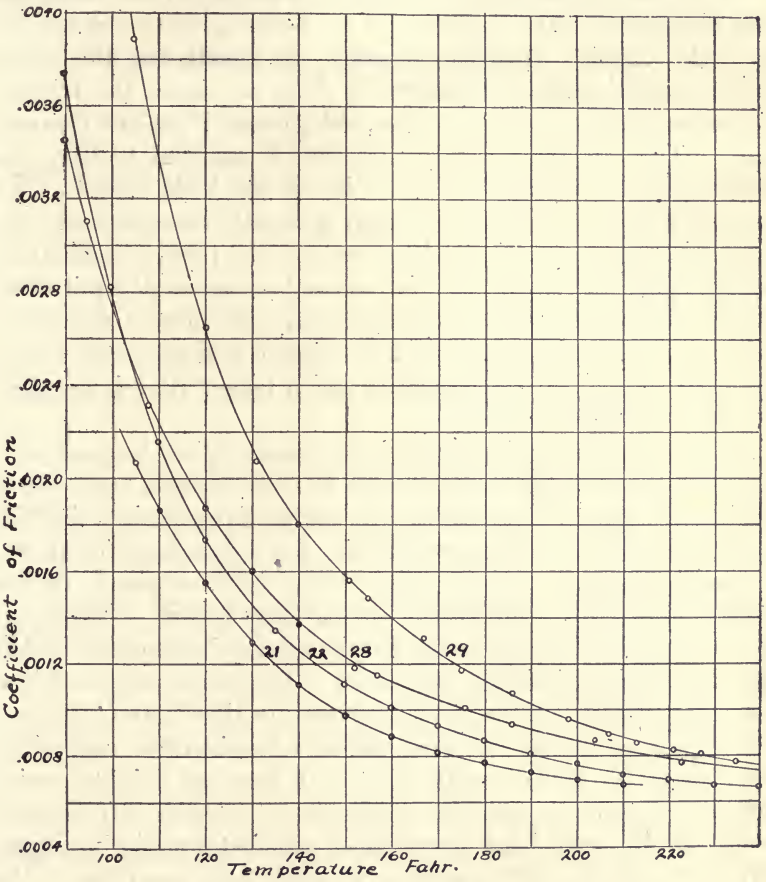

FIG. 37.—FRICTION TESTS OF ENGINE OILS.
PRESSURE, 170 POUNDS PER SQUARE INCH. SPEED, 94 FEET PER MINUTE.

the best results have been obtained by the use of a hardened and polished steel journal $\frac{1}{4}$ inch in diameter, running between two brass bearings about 1 inch long; on this small journal pressures up to 8,000 pounds per square inch may be applied if necessary. The samples of oil to be compared are contained in small brass

* *Journal of the American Society Naval Engineers*, 1897.

cups placed inside the case and surrounding the test journal, each cup having a wire for transferring oil to the journal; the case, samples, and journal are together heated to any desired temperature.

13. In testing for body, the oils are compared in pairs, being applied alternately at the upper end of the bearing; one being applied until the friction becomes constant or nearly so, the other is then applied until it displaces the first, and again the friction becomes constant at the new value; this process is repeated several times. The oil giving the less friction is assumed to have the greater body. In this way the order of the body values of six samples of oils of the same class may generally be determined for any given temperature in an hour or less; the friction indications rapidly follow the changes of the oils and are generally quite consistent. When the oils to be compared are of different classes (as mineral oils with fixed oils), the first friction indications on changing oils are frequently misleading, and a longer time is required to insure certainty of results.

14. The speeds for the body tests are made rather low and pressures not unnecessarily high, in order to avoid heating and wear of the journal, since it is essential for comparative purposes that the surfaces should be in the same condition for both samples compared —a requisite which above all others led to the development of this method of testing. Again, the actual temperature of the oil at the test surfaces is shown more nearly by the thermometer if but little heating by friction be permitted. The writer has not found the order of body values, as determined by this method, to vary with the speed or the pressure within a considerable range. A speed of 50 to 100 revolutions (3 to 6 feet per minute), with sufficient pressure to make the coefficient of friction only as great as .01 to .03, have been found most satisfactory; the pressures being from 500 to 5,000 pounds per square inch, according to the character of the oils.

15. The results of this method of testing for body agree thoroughly with the principal fact hitherto established with regard to body as distinguished from viscosity—namely, that the mineral oils as a class have much less body than the animal and vegetable oils. For example, in a body test of mineral oil and lard oil having viscosities 98.9 and 83.7 respectively at 90 degrees F., as determined by the Dudley pipette, the lard oil, although the less viscous, gives very decided evidence of greater body, by its much

smaller friction. On the other hand, certain cylinder oils, wholly or largely mineral, and exceeding lard oil greatly in viscosity have also greater body than lard oil.

16. The method of testing readily indicates differences in body in samples of nominally the same oil from different manufac-

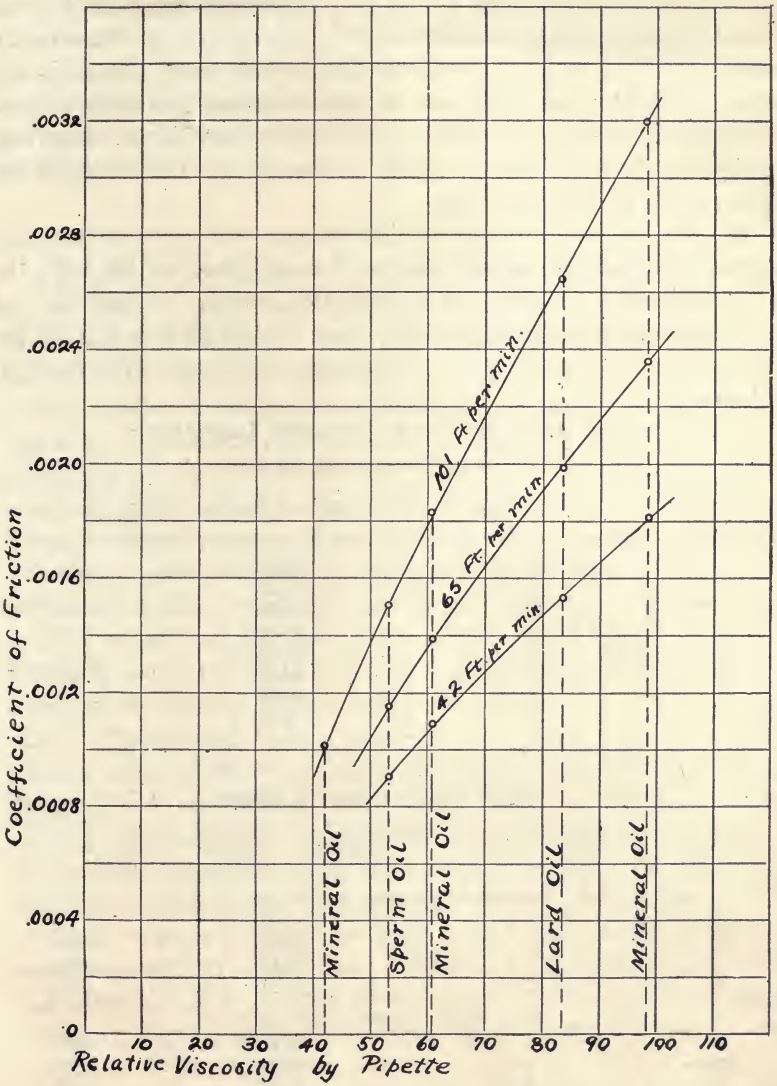

FIG. 38—FRICTION, VISCOSITY, AND SPEED.
PRESSURE, 340 POUNDS PER SQUARE INCH. TEMPERATURE, 90 DEGREES FAHR.

turers, such as lard oil; change of body in a given fixed oil on exposure to the air for some time; the addition of small proportions of mineral oil to animal or to vegetable oil, or vice versa.

17. Among the specific results thus far obtained, the four cylinder oils whose variations of viscosity with temperature are shown in Fig. 36, were found to have body in the same order as viscosity; the body being tested at 180 degrees and 312 degrees Fahr. Similar results were found for the oils in Fig. 37. Again, samples of castor, lard, olive, and sperm oils were placed in the order named by the body test, which order was the same as that of their viscosities. It thus appears that body is in some way related to viscosity, but the relation must be quite different in the mineral oils and the fixed oils.

18. The following tables of observations will serve to show the character of the numerical results of tests made in this way, the oils compared being in the first case two samples of lard oil, and in the second case two engine oils, Nos. 28 and 29 (see Fig. 37 for viscosity tests); speed, 105 revolutions per minute (6.9 feet per minute):

BODY TESTS OF WINTER STRAINED LARD OILS.

Load, 300 lbs.; pressure 2,550. lbs. sq. in.

Oil No.	Torsion, Degrees.	Coefficient of Friction.	Temperature Fahr.
1	33	.0233	72
2	25	.0176	....
1	31	.0218	....
2	24	.0169	....
1	30.5	.0215	....
2	23.5	.0166	....
1	29.5	.0208	....
2	23	.0162	72

BODY TESTS OF ENGINE OILS.

Oil No.	Torsion, Degrees.	Coefficient of Friction.	Temperature Fahr.
	Load, 200 lbs.; pressure, 1,700 lbs. sq. in.		
29	11.5	.0124	69
28	15	.0161	69
29	11	.0118	69
28	15	.0161	69
	Load, 100 lbs.; pressure, 850 lbs. sq. in.		
28	8.5	.0182	120
29	4.0	.0086	120
28	5.0	.0107	120
29	3.5	.0075	120

Thus, while the coefficient of friction is not always constant for any one oil, the one effecting the greater reduction of friction is readily distinguished, and, hence, is to be regarded as having the greater body. This appears to be the principal use which can be made of the numerical results, since the mean values of the coefficients of friction are not characteristic of the oils, but depend also upon the varying degree of roughness of the surfaces, the loads, the speeds, the temperatures, and the kinds of metals forming the journal and the bearing. Nevertheless, great or small differences in the values of the coefficients of friction must be taken as indicating correspondingly great or small differences in body.

DISCUSSION.

Prof. R. H. Thurston.—Professor Kingsbury's paper impresses me as being a very exceptionally valuable contribution to the literature of its subject. The results of his investigation, as illustrated by the diagrams presented, have this very important peculiarity: that they exhibit, as, I think, never has been done before, the laws of variation of frictional resistance when the lubricant is maintained in an invariable state, and in such manner as to show precisely what are the effects upon thoroughly lubricated surfaces, of simple change of temperature and of simple variation of pressure, uncomplicated by variations of condition of the metal surfaces between which the lubricants are placed. These curves are the most complete and the smoothest, on the whole, that I have yet met with, with perhaps the single exception of some of those of Woodbury, representing his experiments on the mineral oils.* These later tests, however, have greater range, and are also, in part, outside the limit of the earlier work. These extraordinarily low coefficients, also would seem to indicate a more perfect attainment of the condition of maximum effectiveness of lubrication—complete separation of the metals, the film of oil being maintained intact throughout. In all earlier experiments, we find evidence of more or less metallic contact and rubbing at some stage of the experiment, converting fluid into solid friction and introducing a change of the law of variation, thus giving curves of correspondingly varying law. Such changes of con-

* "Friction and Lost Work in Machinery and Mill Work," R. H. Thurston, New York, J. Wiley & Sons; London, Chapman & Hall.

dition may be found illustrated in ample extent and variety in work published by me when pioneering in this field.*

The point of minimum friction, to which I then called attention, is better exhibited in Fig. 34 of the paper than in any of my curves; while the latter seem to indicate that this critical temperature of maximum effectiveness of the lubricant is approximated in a variety of oils, organic as well as inorganic. But it is to be noted that journals in ordinary work do not usually attain that temperature and that the difference among the oils increase as the temperature is lower, at least within the allowable range.† The minimum coefficient reported in these experiments is the lowest with which I have met, and may perhaps be accepted as that to which perfect lubrication approximates as the condition of the journal and the system of lubrication approximates perfection.

The very closely hyperbolic form of these curves is another very interesting result of this work. It follows that, since the product of the coefficient of friction and the pressure is, under such circumstances, practically constant, the total frictional resistance, under similarly perfect conditions, is unaffected by change of load on the journal. This is one of those phenomena which conspire to make the friction of the steam engine constant for all loads, as I showed by direct experiment many years ago, in the case, particularly, of the non-condensing engine.‡

Professor Kingsbury's oil-testing machine is a very ingenious and remarkably refined piece of scientific apparatus, and seems especially suited to such peculiarly fine work of investigation as is here illustrated. The difficulties of determining the effect of varying temperature in this case can only be fully understood by those who have attempted such researches, and they seem to be, by this apparatus, admirably disposed of. It brings out, as never was done before, the differences between "viscosity" and what its inventor calls "body," a quality which remains to be defined.

In illustration of the correspondences and the differences between the work of which the paper is descriptive, and that which is ordinarily performed by the standard methods and ordinarily satisfactory apparatus of the laboratory in commercial investiga-

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* Ibidem. † Ibidem.

‡ "Manual of the Steam Engine," vol. ii., chap. v., §§ 132-136.

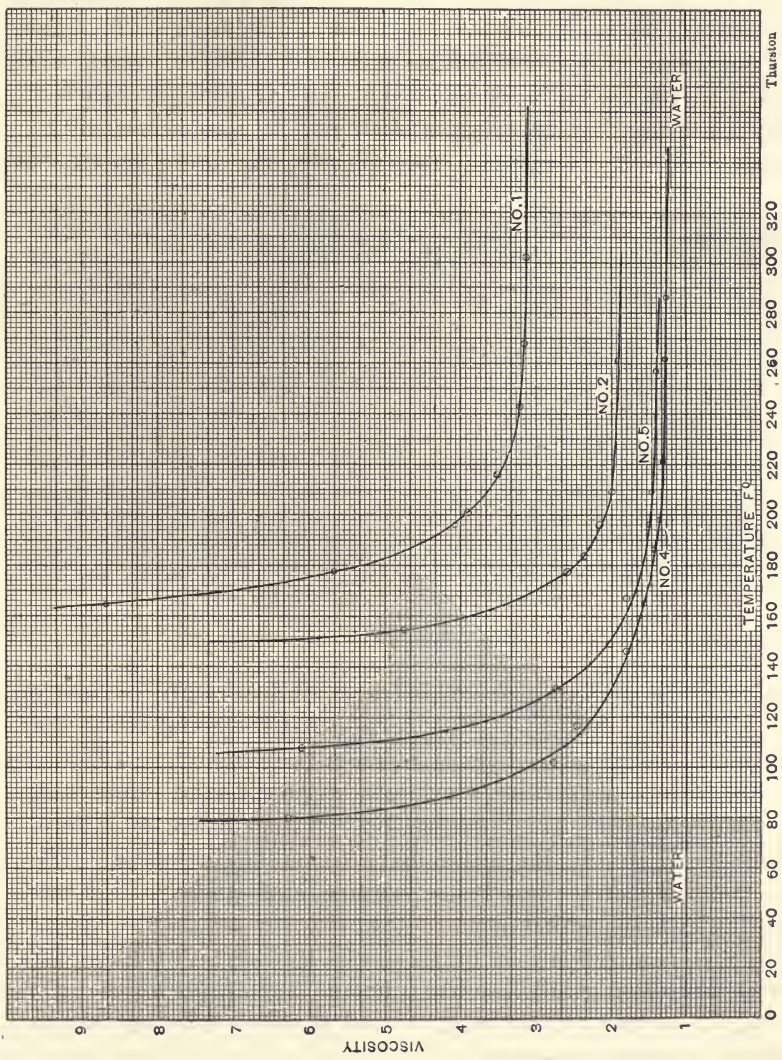

Thurston
NO.1
NO.2
NO.5
NO.4
WATER
WATER
TEMPERATURE F°
VISCOSITY
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
300
320
1
2
3
4
5
6
7
8
9

FIG. 39.

tions for practical purposes, we may compare the diagrams of the paper with those of Figs. 39–41. These are not presented as representing an effort to secure such refinement as would characterize the more deliberate and minutely studied work of research, either in experimentation or deduction, but simply as bringing out the more clearly the difference between ordinary tests and those here discussed, in which every effort is made, and successfully, to insure that the results shall not be complicated by the presence of more than the one variable, the law of which is

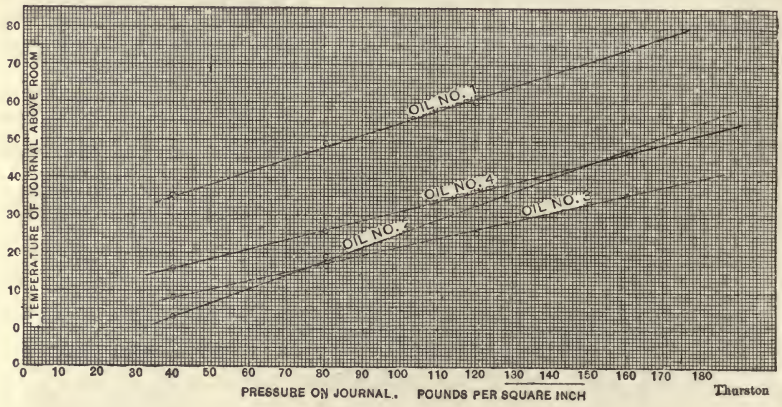


FIG. 40.

sought to be determined. The correspondences are evident on inspection, and it is interesting to see that the commerical work admirably sustains the deductions from the research.

Fig. 39 represents the variation of viscosity with temperature in comparisons of the characteristics of four distinctly different oils: No. 1, an oils old in the market as a "cylinder oil"; No. 2, a "crusher oil"; No. 3, an "engine oil," and No. 4, a moderate-priced "machine oil."

Fig. 40 is a comparison of these oils to ascertain their normal temperatures of steady operation under varying pressures.

Fig. 41 exhibits their relative and absolute friction-reducing values at various pressures.

It is seen that the variation of viscosity, determined in the usual manner, as temperatures change, follow the same general law, and the higher the grade, as rated for the market, the greater the viscosity, *except* that Nos. 3 and 4 are transposed, indicating, possibly, that No. 3 is incorrectly rated and that its

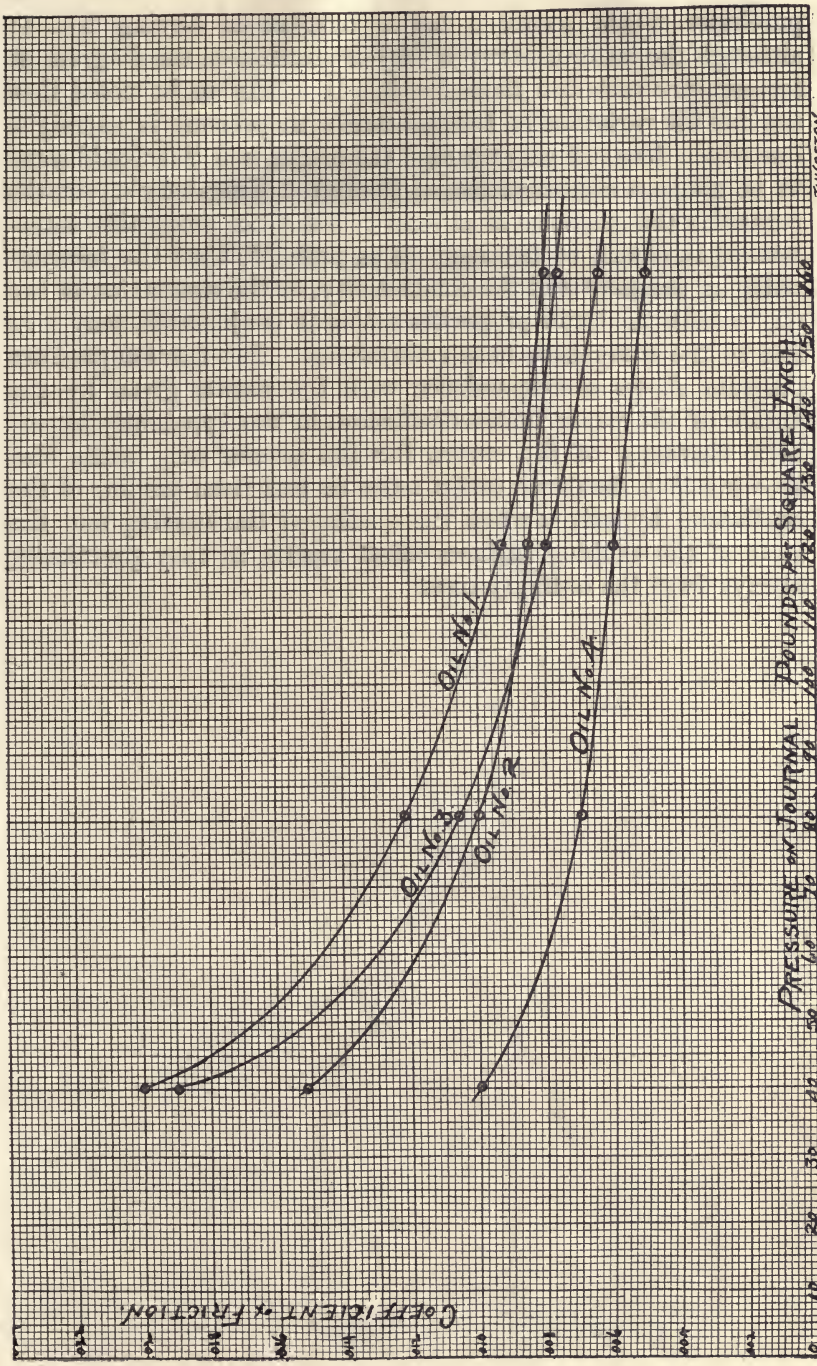

COEFFICIENT OF FRICTION
PRESSURE on JOURNAL. POUNDS per SQUARE INCH.
OIL No. 1
OIL No. 2
OIL No. 3
THURSTON.

FIG. 41.

purchaser would secure a better bargain in buying No. 4; although the latter assumes a higher position on the diagram of temperature variation than No. 3. It is also to be noted that the line for variation of temperature with pressure is a straight line which does not pass through the origin, and this would seem to be another corroboration of Professor Kingsbury's remarks regarding the existence of at least two distinct properties, determining the work of friction. It is probable that the friction-reducing property and the conductivity, or the convection of the oil, or both the latter properties acting in conjunction, may give rise to the differences here referred to. The friction-pressure diagram shows correspondence with the results given by the author of the paper under discussion, in the apparently hyperbolic character of the curves for the several oils, while also showing that the curve varies more or less from the line of the equilateral hyperbola when the quality of the bearing and the state of the rubbing surfaces affect results, as they practically always do in the ordinary tests.

No. 1048.*

THE BURNING OF TOWN REFUSE,

WITH SPECIAL REFERENCE TO THE DESTRUCTORS AT BRUSSELS, WEST HARTLEPOOL, MOSS SIDE, AND WESTMINSTER.

BY GEORGE WATSON, LEEDS, ENGLAND.

Member, Institution of Mechanical Engineers.

1. *Refuse in Great Britain.*—Town refuse in Great Britain varies considerably in character according to local conditions, but it is a fairly safe generalization to say that it consists of one-third by weight of water, one-third combustible matter, and one-third incombustible. The last is withdrawn at the end of the burning process in the form of hard clinker.

The combustible material is largely vegetable and putrescible matter, but it includes, in the United Kingdom, varying quantities of unburned cinder from the wasteful open fires so dear to the English people in more senses than one. This cinder is much more plentiful in winter than in summer.

2. *Continental Refuse.*—Cinder is almost entirely absent from town refuse on the Continent of Europe, where closed stoves are used, producing a fine incombustible ash. The author has frequently proposed that the fine ash, which requires no treatment, should be collected in separate bins in each house, and not taken to the destructor at all.

3. *Collection.*—The nature of the material is affected largely by method of collection, which varies greatly in different towns. For instance, in Edinburgh there is a daily collection, the inhabitants being obliged to place their refuse on the street in receptacles which they provide themselves. These receptacles are usually anything but effective, and the contents are spread about by the wind and the rag-pickers, and mixed with the street sand; sometimes the same cart collects both street sweepings and refuse. The result is

* Presented at the Chicago meeting, May and June 1904, of the American Society of Mechanical Engineers, and forming part of Volume XXV. of the *Transactions.*

a very light sandy material, difficult to burn, which measures no less than 80 cubic feet to the ton.

In Bradford, Leeds and Sheffield, on the other hand, many large ash-pits are used, in some cases combined with privies, and they are only cleaned out at intervals of weeks or months. At one destructor in Bradford the refuse contains 40 per cent. of night-soil. Under such conditions the refuse is wet and heavy, measuring only 40 cubic feet to the ton.

In the author's opinion the prevailing custom of reckoning the amount of refuse burned by weight is misleading, and better comparisons as to the labor involved would be got by reckoning in cubic yards. It is obvious that a ton of bulky refuse requires much more labor in handling than a ton of such wet and heavy stuff as is collected at Bradford.

4. *Refuse as Fuel.*—Town muck is so heterogeneous that it is almost impossible to compare it with other fuels by any of the ordinary methods, and, in particular, calorimeter tests have always seemed to the author to be quite futile. It would appear impracticable to obtain a fair sample small enough to go into a calorimeter of a material comprising garbage, vegetable refuse, dust, straw, paper, rags, bones, broken glass, tin cans, wood, cinders, sacks, old boots, buckets, water-cans, casks, carpets and huge rolls of kamptulicon, to say nothing of the carcases of cattle, dogs, cats and pigs, the last being sometimes brought in numbers after an outbreak of swine fever. Amongst the articles gravely registered as having been destroyed at one of the early Leeds destructors was "one sea-serpent." In two instances parcels of explosives have found their way into furnaces, and blown out the fronts, fortunately without causing bodily injury in either case. To state the nature of the miscellaneous rubbish comprised in the refuse of towns is to demonstrate at once its unsuitability for agricultural manure, and the danger of allowing it to accumulate.

5. *Refuse as Manure.*—No farmer cares to cover his fields with broken glass and tin cans, and in the few instances, such as at Manchester and Glasgow, where some of the rubbish is taken by farmers, it has to be first sorted and ground, and then transported long distances free of charge by the Corporation at a cost exceeding that of destruction by fire.

6. *Danger to Health.*—Accumulations of this material involve the twofold danger of lowered vitality owing to an impure atmosphere, and direct propagation of disease through poisonous germs

carried by flies, rats, dust particles, running water and other agencies. Refuse heaps also frequently occasion great nuisance and expense by spontaneous combustion. "Destruction" of refuse is a somewhat scientific term, but it serves to distinguish the method of disposal by fire from the reduction or digestion processes used in the United States for recovery of grease and other matters, and from the slower process of nature, involved in disposal by dumping, which have been sometimes referred to as the "method of putrefaction."

The author does not propose to go through the history of destructors in Great Britain, where they originated. The late Mr. Alfred Fryer, of Nottingham, was the pioneer, and built his first successful destructor in 1875. It may be thought that nearly thirty years is a long time for the development which has taken place to have occupied. It must be remembered, however, that being dependent upon municipal enterprise, rapid advance has only been possible in recent years.

The author regrets to say that, in spite of the demonstrations made many years ago of the danger of filling old pits and hollows with this material, the practice of doing so, and of afterwards erecting dwelling houses upon it, has prevailed in most towns and cities in spite of all sanitary considerations, and has only given way to the use of destructors when every pit and hollow within the municipal boundary has been filled.

In passing over the history of the development of destructors, the author is also obliged to omit reference to many types which have proved successful, and have served to advance the general practice, and to many names honorably associated with such advance; and in dealing later with the details of certain installations has thought it best to confine himself strictly to his own experience. He will therefore deal only with plants on one particular system; at the same time remarking that there are many other destructors of different types in use which reflect the greatest credit on all concerned.

7. *Natural Draught.*—The earlier destructors were all upon the natural-draught system, with slow combustion, low temperatures, and little or no provision for raising steam. In charging and clinkering the furnaces large doors were kept open for about one-third of the time, and the strong chimney draught above the grates necessarily drew in tons of cold air over the fire through such openings, making high furnace-heats impossible. Frequent

complaints of nuisances from the chimney shafts caused by stinks and dust resulted.

8. *Fume Cremators.*—Fume cremators, introduced by Mr. Jones, the borough engineer of Ealing, were added in many cases, consisting of secondary fires in the flues fed with coal or coke, over which the products of combustion had to pass, thus scorching them and largely obviating the nuisance. These fume cremators certainly rendered a continuance of working possible in many cases, but owing to the high cost for fuel they were not always regularly worked.

9. *Maintenance.*—One advantage of natural draught and low temperatures was that the brickwork was easily maintained, and furnaces of the most ordinary construction have been found to last fifteen or sixteen years without much expense in renewing the fire-brick linings. Even to-day engineers are found who advocate low temperatures on this account.

10. *High Temperatures.*—Modern plants working at high temperatures with forced draught, and with steam-raising as one of the most important objects, involve much more difficult problems as regards construction and maintenance, and also as regards the protection of the stokers from heat and back-draught. No ironwork can last long in a modern destructor unless cooled in some special manner, and contrivances such as dampers within the furnace for closing or partly closing the draught outlet when the furnace door is opened, have become not only superfluous but impracticable. The author has learned to look with the greatest distrust upon any supposed improvement involving the introduction of unprotected ironwork into the hot furnaces or flues.

11. *Storage of Heat.*—In high-temperature furnaces heat is stored in the brickwork to such an extent that the furnace arch hardly ceases to glow even when nearly a ton of cold refuse has been freshly charged, and consequently the temperature does not fall below the point, say 1,250 degrees Fahr., at which septic gases might be given off.

12. *Plenum System.*—No one has contributed more to the attainment of such temperatures than Mr. William Horsfall, of Leeds, who first introduced the plenum system and the front exhaust flue. Recognizing that the principal cause of low temperatures in the old natural-draught furnaces was the inrush of a large excess of cold air over the fire during the lengthy processes of charging and clinkering, he set himself to apply forced draught

on the closed ash-pit system, first by means of fans and later by steam blast, and this enabled him to choke the furnace outlet in such a manner as to always maintain a slight pressure above the atmosphere within the furnace, thus absolutely preventing the admission of cold air when the furnace doors were open and allowing loose fitting doors to be used.

13. *Front Exhaust.*—He placed his exhaust flue in the front portion of the furnace arch, thus reversing the draught and bringing all the fumes given off by the refuse drying on the hearth over the hot fire and "cremating" them effectually within the furnace itself.

14. *Steam Raising.*—An important result of the introduction of high-temperature destructors has been that town refuse has now come to be regarded in some quarters as fuel. When it is remembered that the combustible matter in the charge is, say, only one-third by weight of the whole, the claims which have been put forward of a commercial evaporation of three pounds of water to high-pressure steam per pound of muck can only be regarded as ridiculous, for they would place the combustible portion of the material on a level with best Welsh coal. It is to be regretted that inventors have come forward from time to time offerings guarantees of such results in regular working. Their claims can only be accounted for by their lack of experience. Failure in the fulfilment of such guarantees under contract have led to a serious reaction, and there is now a tendency to discredit altogether the possibility of steam-raising from town refuse. As usual the truth is found between the two extremes.

15. *Steam as a By-Product.*—The great value of steam as a by-product of a destructor plant designed primarily for the disposal of the muck (but well arranged for the production and utilization of the steam) has been amply demonstrated. It may now be laid down with perfect confidence that, on an average, town refuse in Great Britain may be expected to evaporate in every-day working its own weight of water from and at 212 degrees Fahr. In many places it is safe to guarantee 1 pound of steam per 1 pound of refuse in summer and $1\frac{1}{4}$ pounds in winter. As much as 1.5 pounds may be got on test with careful management, as will be seen in the record of a test at West Hartlepool, given in Appendix 1.

The value of the refuse as fuel varies considerably in different districts, and according to the season and the amount of coal used

by each household. In some colliery districts the miners are provided with coal free of charge, and, as might be expected, the best steam-raising refuse is to be met with there.

On the other hand, the author has had experience of seaside towns, such as Lowestoft and Ramsgate, where in hot weather hardly any coal fires are found (gas-stoves being used for cooking); and the refuse consists largely of garden rubbish, garbage, and bad fish—poor stuff for steam-raising.

In wood-burning countries, and where closed stoves are used, not only does the absence of cinder mean a very poor fuel, but the fine ash (if collected with the refuse) has a tendency to surround and choke such combustibles as may be present, and to put out the fire altogether.

16. *Refuse Auto-Combustible.*—Nevertheless, the author has had, up to the present time, no experience of town refuse which was not auto-combustible, except that from the eastern district (the poorest part) of Berlin. Even there he has by no means given up the problem of burning the refuse by itself. Monte Carlo and Pernambuco are places where little or no cinder is found in the muck, and yet destructors are in successful operation at each of these without the addition of other fuel.

17. *High Pressures of Steam.*—To revert to the conditions prevailing in Great Britain, it used to be said that high pressures of steam could not be obtained from the heat of destructors; but a consideration of the temperature of the gases leaving the destructor, which varies from 1,700 degrees Fahr. to 2,000 degrees Fahr., might have shown the fallacy. As a matter of fact, steam is being obtained at Accrington (with a "Lancashire" boiler) and Moss Side, Manchester (with "Water-Tube" boilers), at 200 pounds per square inch, while pressures of 120 pounds per square inch are now quite common.

The conditions for the best working in the destruction of refuse are identical with those required for efficient steam-raising; and with high temperatures the clinker is harder, the working more rapid, and the chimney clearer of smoke.

18. *Arrangement of Cells and Boilers.*—The grouping of the cells and boilers is an important and much debated subject. Many early designers, attaching great importance to the radiations from the burning mass, held that the heating surfaces of the boiler should be immediately over the fire, and as close to it as possible. It is almost needless to say that the results were disappointing,

as the temperature necessary for complete combustion was never reached at all, owing to the cooling effect of the surfaces of the boiler.

Later developments of the same idea are still in evidence in the "sandwiching" of boilers and cells, that is, the placing of a cell on each side of a single boiler and delivering the products of combustion through side openings in the cells directly below the boiler tubes. This plan is much more satisfactory than the earlier one; but the chief drawback to its adoption is that when either of the cells is newly charged, and the gases may be escaping at a temperature below what is necessary for complete combustion, they may come into contact with the cool surfaces of the boiler unconsumed, and so pass away to the chimney, while fluctuations of steam pressure may also occur.

In the author's opinion, the best grouping of cells and boilers is to arrange the cells in blocks or batteries not exceeding six or eight in number, and to place the boiler or boilers as near as possible to the block of cells.

In the case of the back-to-back cells, arranged in such a block with the main flue underneath them, there is little or no loss by radiation, as the main flue is surrounded by cells. At the same time the immense advantage is secured that the products of combustion from all the cells are thoroughly mixed in the red-hot flue before passing to the boiler, and thus if one of the cells happens to be somewhat cooler than the rest, any unburned gases that may escape from it are thoroughly burned in the main flue, where there is always some excess of air. All the heat generated in the cells is carried forward by the hot gases, except the small proportion which escapes by radiation from the outer surfaces of the furnaces themselves.

With batteries of cells in a single row, Fig. 515, it is also possible to arrange that the loss by radiation shall be small. A further advantage is that the cells, being close together, the stokers have less floor space to work over, and the arrangement of railways or conveyors for removal of clinkers is simpler than with boilers and cells alternately.

19. *Working in Rotation.*—Under good management the working of the cells is kept strictly in rotation, a time-table and a clock being provided, and good time-keeping in charging and clinkering being insisted upon.

20. *Constant Pressure of Steam.*—With such methods the tem-

perature of the gases entering the boilers may be maintained almost constant, and the old complaint of the steam pressure

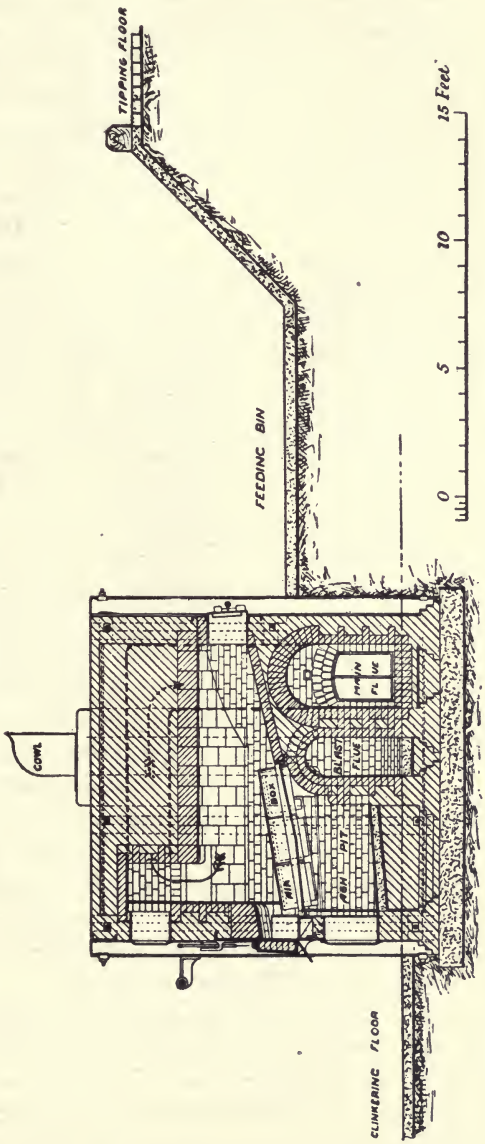


FIG. 515.—SINGLE ROW BACK-FED DESTRUCTOR FURNACE.

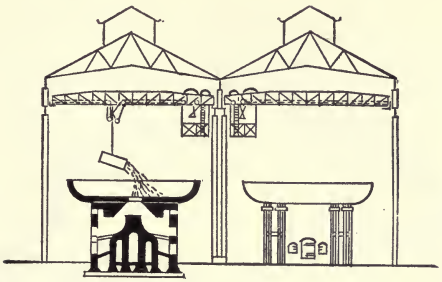
fluctuating disappears entirely. The clinkering and charging of each cell takes place every one and a half to two hours, according to the nature of the material. In some cases clinkering every hour

has been tried, but the experience of the author is that a longer run gives a harder clinker, and a better result all round, any small loss of steaming power being more than compensated by reduced labor and more perfect combustion. The capacity per cell depends mainly upon the grate area, and the strength of blast, but also, of course, very greatly upon the attention of the firemen.

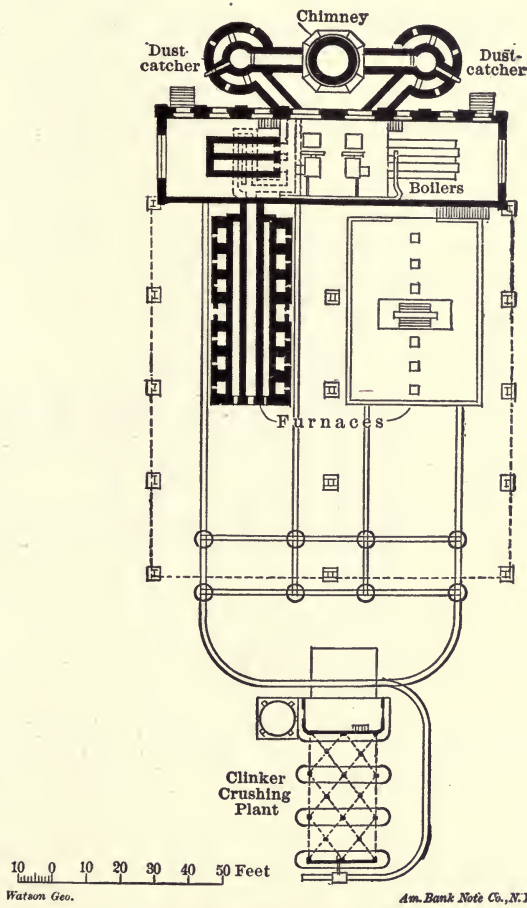
21. *Blast.*—With refuse for fuel it is extraordinary how soon a high-pressure blast blows the fires into holes. With forced draught on the closed ash-pit system, a few minutes' use of the rake to fill up the blow-holes will increase the draught gauge from one-quarter inch to over an inch of water column. In some cases a draught of as much as 2 inches is used, but there are two great disadvantages to such pressures: first, that much more labor is required (almost constant trimming of the fires being necessary to fill the blow-holes), and, second, that the high blast sends up quantities of hot dust and sparks, which cake on the roof of the furnace and in the flues, necessitating constant cleaning.

22. *Grate Area.*—From 25 to 30 square feet of grate area has usually been provided in each cell up to recent years; but, of late, cells having 42 square feet of grate area have become common; and also much smaller cells are being introduced for special purposes, such as for use in hospitals, asylums and factories. For plants up to four cells furnaces of 30 square feet of grate area are most suitable; but for larger plants furnaces of 42 square feet are preferable, as the use of larger cells reduces the cost of labor. The length of the grate is usually 6 feet from back to front in both sizes of cells. It is customary to work throughout the twenty-four hours for six days a week; but in some cases, where steam is only required for night work, such as in electric lighting, the hours of working have been reduced, and the number of cells and boilers correspondingly increased. It is quite possible to bank the fires for twenty-four hours, and even longer, so that they do not require relighting on a Monday morning.

For the purpose of illustrating his remarks, the author has provided drawings and photographs showing several different types of plant in connection with the construction of which he has recently been employed. The installations chosen for this purpose are those at Brussels (twenty-four cells, Fig. 516), West Hartlepool (twelve cells, Fig. 517), Moss Side, Manchester (six cells, Fig. 518 and Fig. 519), and Westminster (six cells, Fig. 520 and Figs. 521 and 522). Each of these destructors is placed in a



Transverse Section.



Plan.

FIG. 516.—DESTRUCTOR OF 24 CELLS.

Grate area, 30 sq. feet each. City of Brussels.

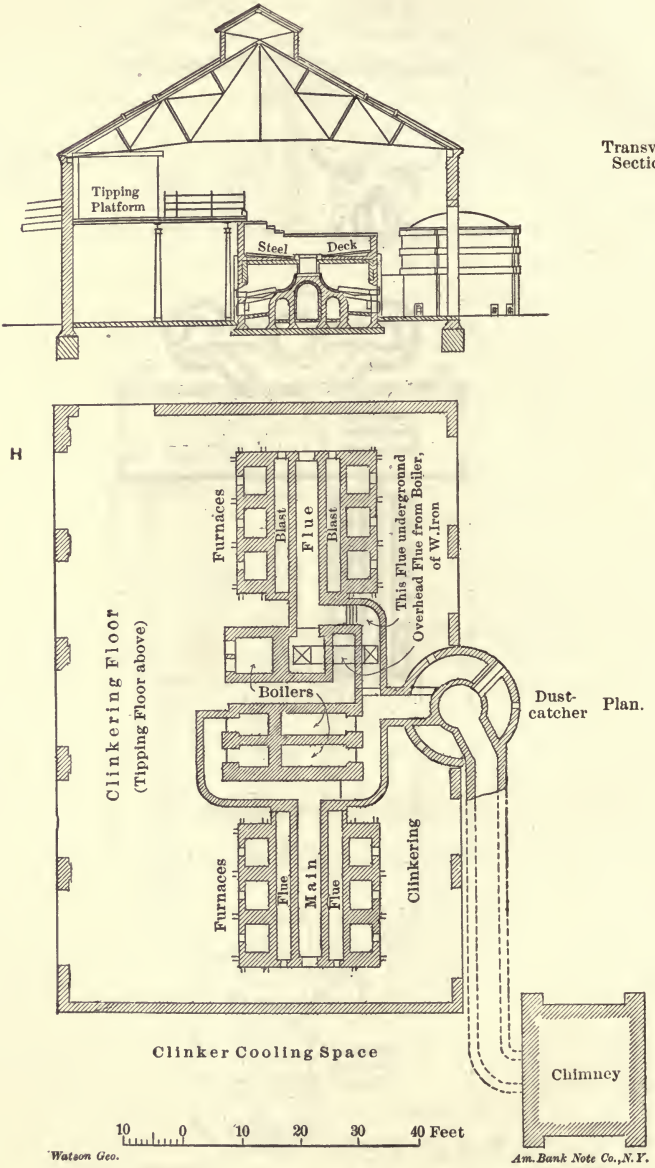


FIG. 517.—DESTRUCTOR OF 12 CELLS.

Grate area of 30 sq. ft. each. Corporation of West Hartlepool.

densely populated district, and all are in full operation without causing any inconvenience or complaint.

23. *Destructors at Brussels and West Hartlepool.*—The Brussels and West Hartlepool furnaces, Figs. 516 and 517, are of the back-to-back type, fed by hand through holes in the deck on the top of the furnaces. These feed holes are of somewhat peculiar construc-

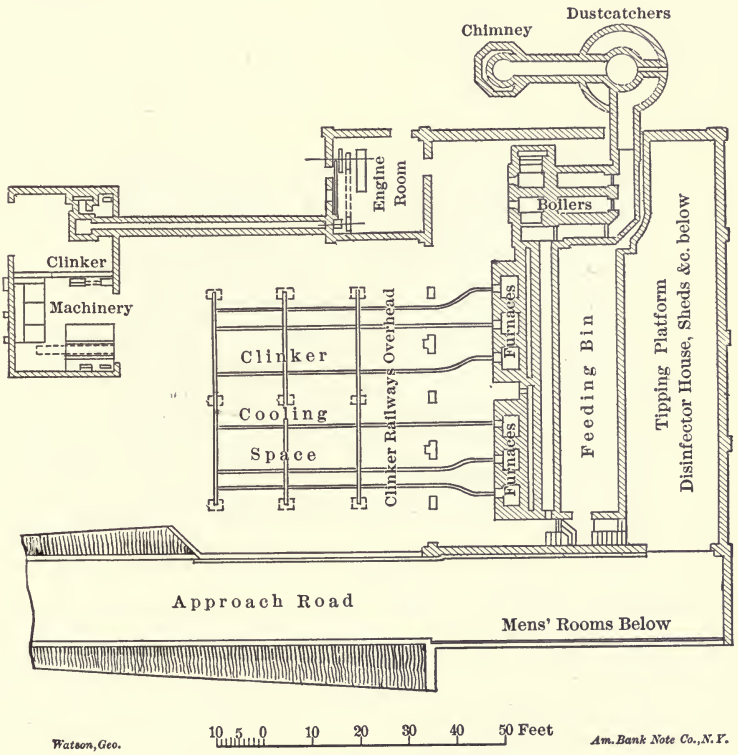


FIG. 518.—DESTRUCTOR OF 6 CELLS.
Grate area of 30 sq. ft. Moss Side, Manchester.

tion; they are so designed that no lid is required, the refuse itself being trodden down into the holes to stop them up. It will readily be seen that the use of a lid on a deck littered with rubbish would be very inconvenient. In order that the method of stopping above referred to may be used, there is provided immediately below the feed-hole (which is common to two furnaces) a flat table or saddle so arranged that when refuse is dragged or shovelled into the hole and allowed to lie on the saddle it fills up and chokes the opening. When it is desired to charge either furnace the refuse

is simply pushed off the saddle with a prong—a very simple operation, requiring very little labor.

The deck is 6 feet below the level of the tipping floor, thus forming a convenient bunker for the reception and storage of the muck as near as possible to the feed-holes. In the West Hartlepool plant the deck is formed of steel plate separated from the top of the furnace by an air space in order that it may be kept cool; while at Brussels the deck is carried entirely by the furnaces and is formed of reinforced concrete. At West Hartlepool the refuse is carted up an inclined way with a gradient of one in twenty,

FIG. 519.—FURNACES. 6-CELL PLANT. MOSS SIDE.

while at Brussels there are two electric overhead travelling cranes, one over each group of twelve cells, which lift the cart-bodies off the wheels and dump their contents on to the decks. The latter system has also been adopted at Hamburg, and in the plant now being completed at Zürich.

24. *Moss Side Destructor.*—At Moss Side, as will be seen on reference to the drawing, Fig. 518 and Fig. 519, the six cells are arranged side by side in a single battery with the two water-tube boilers at the end of the battery, the charging openings are at the back of the furnaces, and are provided with suitable furnace-doors. These doors are now generally planed to fit their frames, and are lined with fire-bricks. They are raised and lowered by levers and balance weights, and give a good command of the furnaces, which slope downwards to the front.

The refuse is tipped from the carts into a bunker of concrete, and having a sloping back, so that the toe of the heap of refuse is

never far from the surface mouth, into which it is thrown by a shovel. This method would seem at first sight to involve more labor than the top feeding, but it is found in practice that, while it costs more to put in the muck with a shovel, the labor in front of the furnace in dragging forward and trimming is correspond-

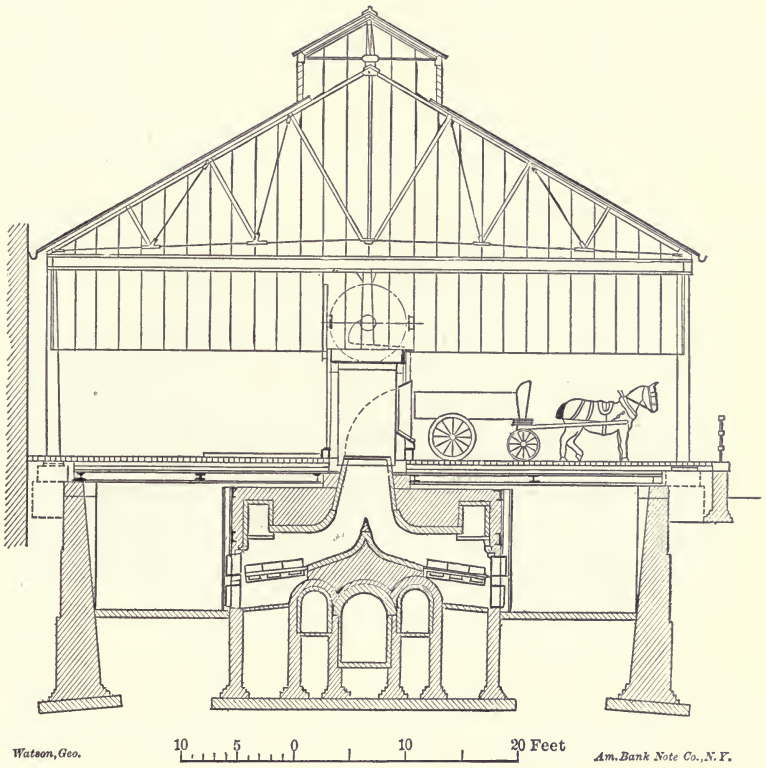


FIG. 520.—DESTRUCTOR OF 6 CELLS.
Grate area of 42 sq. ft. each. City of Westminster.

ingly lessened. It may be remarked, however, that even at West Hartlepool, where the refuse is not bulky, the labor of charging costs one-third of the stokers' wages, and it is, therefore, a matter of the utmost importance to reduce it as much as possible.

25. *Westminster Destructor.*—The Westminster Destructor, shown in Fig. 520 and Figs. 521 and 522, has been erected on the Shot Tower Wharf, at the south side of Waterloo Bridge, in the center of the metropolis, and serves a district of which Covent Garden Market and the Strand form important features. The

refuse is exceedingly bulky, consisting largely of market and shop refuse, paper, packings, straw, and all kinds of light material, measuring not less than 90 cubic feet to the ton. To feed such stuff by means of a shovel would be very costly, and therefore special means were demanded. The conditions have been met by a new design of furnace, into which the four-wheeled collecting carts and steam motor-wagons tip their loads direct, no storage being provided other than the spare carts which are brought in from the yard and discharged as required. Sufficient spare wagons are provided to keep the destructor in full operation day and night. The furnaces are six in number, each having 42 square feet of

FIG. 521.—WESTMINSTER DESTRUCTOR. EXTERIOR.

grate area. They are arranged back to back, with a water-tube boiler at the end of the battery. A very large feed hole, 6 feet by 4 feet, is common to two furnaces, there being thus three holes in all.

The carts can be brought to either side of the feed-holes to dump. Great difficulty was experienced at first in arriving at the correct shape for the feed-holes—that is, expanding downwards, as now shown, so that the refuse cannot arch over in the furnace mouth. A worse difficulty was that of getting such large lids as were required to be quite smoketight when closed. Metal to metal joints and grooves filled with sand were in turn found unsatisfactory, and the handling of such heavy lids was also difficult. Eventually hinged lids, counterbalanced by weights hung from volute quadrants, so as to be perfectly balanced at all positions, were adopted.

Each lid dips into a water-seal which surrounds the furnace mouth, the water, which evaporates slowly, being maintained at the proper level by a ball cock. This makes a perfectly gas-tight joint, and not only is the trouble of smoke from the feed-holes obviated, but it is found that the absolute tightness of the joint, preventing even a small leakage, causes the furnace gases to pond up beneath the lid and to remain stagnant, thus forming a shield of comparatively cool gas, which protects the lids from the furnace

FIG. 522.—CART TIPPING, WESTMINSTER.

heat. As long as there was any leakage at all, the heated gases drew that way, and kept the furnace mouths and lids red hot, which caused them to crack and give trouble.

A hopper, constructed of wrought-iron plates and set on hinges in a position at right angles to that of the lid, is lowered over the opening, so as to prevent refuse getting into the water-seal. This hopper is balanced in a similar manner to that described for the lids, and both hopper and lids are easily worked by hand by means of large chain wheels. The labor of charging is thus saved, the muck is never handled at all, but shot straight into the furnaces. This is an advantage from a sanitary point of view, but it is not universally appreciated, as it puts an effective stop to trade in rags, bones, and dirty glass bottles, which can no longer be picked out by the stokers and sold.

Before leaving the subject of the Westminster destructor, which is the first of its kind, the author wishes to acknowledge his great indebtedness to the Engineers of the City, Mr. Bradley and Mr. Ventris, and to Dr. Priestly, the Medical Officer of the Health of Lambeth (whose duty it was to inspect the working of the plant) for their invaluable aid in overcoming the initial difficulties of the undertaking.

The author has already remarked that he would like to see the duty of destructors reckoned in cubic yards rather than tons, as giving a fairer basis of comparison. It is not, however, in the least likely to come into vogue, since the wagon-weighing machine is such a simple and convenient method of taking the quantities.

26. *Labor in Working.*—The author would, however, suggest a further reform in comparative figures to the effect that the labor bill in working the furnaces should be reckoned, not in pence or shillings per ton (it varies even in England from sevenpence half-penny at Moss Side and elsewhere to two shillings and threepence at several London destructors, for furnacemen only), but in tons dealt with per man per hour. This would eliminate the discrepancies due to the stokers' wages and hours of labor varying so much in different districts. A destructor in which a stoker can deal with a ton per hour during an eight-hour shift may be considered satisfactory, and this is being attained at Westminster. There can be no doubt that a very large saving is effected in charging direct from the carts in the case of a destructor dealing with 72 tons of bulky refuse, such as that of the Strand district, every twenty-four hours.

27. *Mechanical Stokers.*—Many attempts have been made to effect the operations of stoking and clinkering by mechanical grates, but hitherto without much success. Mechanical grates of many kinds have been tried, and large sums sunk in such experiments. The author has had some expensive experience in this direction, and the opinion that he has formed is that any attempt to make the burning process continuous instead of intermittent will fail, unless some entirely fresh method be found. The quality of the refuse as fuel is too poor to enable the fire to creep back through a comparatively thin layer of it as fast as the material must be moved forward to give anything like a reasonable output; and after the mechanism has been set so as to give only about three of four tons per cell per twenty-four hours, it has been found that the speed was too great for the fire, which was very soon

all ejected from the furnace. The fuel also varies so much that one part of the fire will have become black clinker while other parts are insufficiently burned. Trimming and clinkering by hand and the intermittent system of firing in one-and-a-half or two-hour heats at present hold the field.

28. *Clinkering.*—The operation of clinkering is one requiring both strength and skill. The mass of clinker is often 5 inches thick over the whole grate surface, and in breaking it up and withdrawing it from the furnace the stoker must be careful to turn it over so as to throw off any live fire on the top, which he afterwards spreads evenly over the grate for the purpose of lighting the new charge of refuse. The blast, which is shut off during clinkering, should be put on again for a few minutes before charging, so as to prepare a bed of hot fire for the reception of the charge.

Another method adopted with some well-known types of furnace is to feed them continuously from the front by hand and to clinker also continuously from each portion of the grate surface in turn. In furnaces both fed and clinkered at the front, no drying hearth is provided, and there is therefore no preliminary drying of the refuse. This system is adopted by the author for small portable destructors and furnaces for hospitals, but not for larger installations, on account of the higher cost of labor. Some engineers use continuous grates without any division into cells above the grates, the ash-pits only being divided. The author, however, strongly favors the cellular system, which lends itself much more readily to repairs, as one cell can be repaired at a time without stopping the others, thus rendering a duplication of plant quite unnecessary.

29. *Detailed Description of Furnace.*—The different methods of charging the furnaces having been described, it may now be convenient to deal with the details of those parts of the furnaces which are common to the three types at Westminster, West Hartlepool, Brussels and Moss Side respectively. The refuse is first piled up on the drying hearth above the main flue and is afterwards raked from the front on to the grate-bars as may be required. The drying hearth and grate-bars slope down about one in six towards the furnace front; this greatly reduces the labor of working the furnaces. The grate-bars have narrow spaces—about three-sixteenths of an inch—and are made in single lengths of 6 feet, four bars being cast together. This enables the chisel tools used in clinkering to be worked from below along the whole length

of the grate-bars without check. A wide "deadplate" is provided in front, and the furnace mouth is closed by a door extending the full width of the furnace and lined with fire-brick blocks. The furnace door slides upwards to open, the working faces being planed, and the door suspended from a balanced lever. In closing it falls into wedge-shaped catches which force it tight against its frame. Such doors are found much more suitable than hinged doors, which were commonly used in early destructors, as the latter exposed their red-hot inner surfaces to the stoker on being opened.

In each main furnace door is fitted a small rake-door, just large enough to admit the stoker's rake to enable him to trim the fire and pull down the muck without exposing himself to the heat of the furnace. The main door need only be opened for the operation of clinkering. The ash-pits are closed by suitable air-tight doors, and are sloped every way towards the door, to facilitate the removal of ashes.

30. *Fireclays.*—The whole of the interior of the furnace is constructed of specially made fireclay blocks of the best quality, dovetailed together, and all of the arched construction. Flat fireclay lumps over openings and flues are found to crack. Owing to the great heats attained and the frequent changes of temperature, it is necessary to select the fireclay with care, to avoid troubles from expansion and contraction. Bricks too rich in silica expand and contract far too much, and clays which answer well for metallurgical purposes are often found too brittle for destructor work. The fireclays which are found suitable for use in destructor furnaces neither contract nor expand under heat, and are able to withstand frequent changes of temperature, the maximum being about 2,300 degrees Fahr. They should consist of from 60 to 70 per cent. silica and 30 to 40 per cent. alumina. A slight admixture of iron is harmless, but lime, potash and soda, which are usually found in such clays, should only be present in very small quantities, as they tend to act as a flux. The beds of fireclay at Leeds, Sheffield, Glenboig, near Glasgow, and Stourbridge are all of suitable quality. The bricks and blocks must be very truly formed, in order that the thinnest possible joints may be made. It is also important that the fire-bricks should be a little thicker than the common bricks with which they have to bond, because naturally the joints in the common brickwork will be thicker than would be desirable in the fire-brick work.

31. *Fans or Steam-Blast.*—The draught is forced by means of

fans or steam-jets. The question as to which gives best results, dry air or steam-jet blast, is frequently debated. A blast of about one inch of water column is found to answer best with average refuse. The use of higher pressures causes the fire to burn into holes too frequently, and thus causes too great an excess of air to pass, and consequently lowers the temperature.

This pressure can be obtained easily and economically either by steam-jet blast or by a centrifugal fan. For working at a rate of 10 tons per twenty-four hours, on a grate area of 30 square feet, or at a rate, say, 30 pounds of refuse per square foot of grate per hour, a volume of about 700 cubic feet of air per minute at atmospheric pressure and temperature for each cell is required, or, say, 23 cubic feet of air per minute for every square foot or grate area in use. To deliver this quantity of air at the requisite pressure requires, with an efficient steam-blast apparatus, about 100 pounds of steam per hour for each cell of 30 square feet grate area, whereas with a good centrifugal fan only one-fifth of this amount, or, say, 20 pounds of steam per hour, is needed. To compensate for the larger consumption of the steam-jets, it may be stated that they give a higher temperature and evaporation than the dry-air blast, provided that the refuse is rich enough in carbon to give the necessary temperature for the dissociation of the steam, which forms water-gas in the furnace, and which greatly improves the combustion. This action has been explained * by Lord Kelvin and Dr. Archibald Barr in the following words: "A more important function is, however, fulfilled by the steam. In coming into contact with the incandescent fuel it is decomposed, the hydrogen being freed, while the oxygen combines with the carbon in the fuel to form carbon monoxide. This decomposition of the water is effected by heat abstracted from the lower part of the fire, where it can be of comparatively small value for the cremation of the distillate. The 'water-gas' (hydrogen and carbon monoxide) passes upwards to be burned by the excess air which it meets with over the fire, thus serving to increase the temperature, which would otherwise exist at the meeting of the products of combustion with the gases distilled from the raw material."

The importance of the action of the steam-jet was strikingly exemplified in an instance which came under the observation of the author at Bury, Lancashire, where in a new destructor it was not

* Report on the "Horsfall Destructors," 1898.

found possible to reach the guaranteed temperatures in the flues by means of dry-air blast. The substitution of steam-blast for the fans immediately gave and maintained the required temperature. It is fully demonstrated that, provided always the refuse is sufficiently rich in carbon, a steam-blast, although it uses more steam for the draught, yet so increases the amount of the total steam raised as to give a better steaming result on the whole than the fans. With the refuse of Hamburg and Berlin, however, it is doubtful whether the same can be said, because the use of closed stoves in those cities renders the refuse very poor as regards combustible cinders. One hundred pounds of steam used in the blast for each cell of 30 square feet grate area per hour may be considered a very good result for steam-blast, and it can only be secured by using efficient arrangements.

A steam-jet has been devised by Mr. C. W. James, in conjunction with the author, for this purpose, in which the nozzle is flat instead of round, thus yielding a ribbon of steam instead of a plug of steam. It is probable that the air is carried through the blast-tube mainly by means of surface friction between the jet of steam and the surrounding column of air, and it is therefore advantageous to have the greatest possible surface on the steam-jet per pound of steam passing. This is, of course, obtained by making the jet flat and thin. A thickness of one-fortieth of an inch is found best. It is also important that the steam should be superheated, so as to prevent condensation and obstruction in the nozzle itself. A pressure of 30 to 40 pounds to the square inch for the steam-blast gives the most economical results. Reducing-valves are used, giving this pressure whatever may be the working pressure of the boiler.

A further great advantage to be set down to the credit of the steam-blast, as compared with fan draught, is the fact that it protects and lengthens the life of the ironwork exposed to heat in the furnaces, such as, for instance, grate bars and the cast-iron side-boxes which are used for heating the blast. After leaving the nozzle the steam in the blast is condensed, and it reaches these iron parts in the form of particles of water. On striking the hot iron it is immediately re-evaporated into steam, carrying with it a considerable amount of heat, and thus keeping down the temperature of the ironwork. The author has seen grate bars taken out of a furnace after six years' constant use very little the worse for wear, whereas with dry air-blast and high temperatures the grate

bars would require to be renewed after six months to twelve months at the latest.

32. *Cast-iron Furnace Sides.*—The steam-jet trumpets used in the West Hartlepool installation are combined with the cast-iron side-boxes which form the sides of the furnaces above and below the grate-bars. These side-boxes serve the double purpose of heating the blast to a temperature of 400 degrees Fahr. in the ash-pit and of protecting the brick-work at the sides of the furnace from the undermining action of the hot clinker. In furnaces with brick-work sides, at the first level it is found that the clinker fuses to the brick-work, and every time it is removed brings away particles of the wall, thus gradually undermining it, and allowing the crown of the furnace to fall before it is anything like worn out. The side-boxes, which are kept cool by the passage of the cold air and by the cooling action of the steam above described, prevent this action. The air is drawn into the boxes from hoods, something like the hood over a blacksmith's fire, placed over the clinker-door in such a position as to draw off any smoke and dust rising during the operation of clinkering. This improves the ventilation of the stoke-hole.

33. *Firing Tools.*—The firing tools consist of prongs or pushers for charging the furnaces, light and heavy rakes, for pulling down and clinkering respectively, and chisel bars. They are all necessarily long, and any means of making them lighter, without reducing their strength, is worth adopting. It is found a good plan to use a weldless steel tube for rake handles, and to make the chisel bars of solid masons' tool steel in order to get the greatest possible strength.

34. *Boilers.*—In the early destructor plants multitubular boilers of plain cylindrical shape, with fire-tubes about 3 inches or 4 inches in diameter, were frequently used. One of the most serious drawbacks to these was the fact that the dust carried in the gases soon choked up the tubes, which had to be brushed out three or four times in the course of a day, and an even worse defect was the stiffness of the boiler, which did not easily yield to expansion and contraction. A destructor boiler, placed at the end of the main flue, is, of course, really a gas-fired boiler, and it is subject to having the gases switched on or off instantaneously.

It was found that with these multitubular boilers the tubes very soon began to leak at the tube plate; moreover, boilers of such a large diameter as were generally used were not very suitable for

high pressures, unless they were constructed in a very expensive manner. For these reasons such boilers are now only used for very small installations, or where steam-raising is a matter of little importance.

It is essential that a boiler which permits its tubes to expand and contract very freely should be used.

The two types of boiler most favored at the present time in connection with destructors are the Lancashire type and the water-tube type. The former has the advantage of large capacity for storage of hot water and steam, and it is also less susceptible to troubles arising from hard water, but it is found, when using destructor gases, that the flue area is insufficient unless the boiler be made larger than would be usually adopted for the same rate of evaporation when fired in the ordinary way with coal. A deposit of dust in the two flues of the boiler is not a very serious drawback, as the boiler is easily cleaned, and the lower half of the flue is, of course, less valuable as heating surface than the upper half. A well-constructed Lancashire boiler is also able to withstand the effects of expansion and contraction, due to the hot gases being suddenly switched on or off by means of the by-pass damper.

In cases where the destructor has to work continuously, while the power is only required for, say, three or four hours a day (so often in the case in electricity stations) the large capacity of the Lancashire boiler for storage of steam power is very valuable.

The water-tube boiler, however, seems destined to have the preference in most new destructor installations. In the first place, it is specially adapted for high pressures.

Steam is raised very quickly, which is a great advantage in starting the destructor at the beginning of each week, because the forced draught cannot be put into operation until steam has been raised. The water-tube boiler is well adapted for resisting the sudden changes of temperature to which it is subjected in a destructor, and it provides ample area for the passage of the gases in proportion to its heating surface. It has another great advantage in the fact that dust can only lodge on the tops of the tubes (when in proportion to its heating surface. It has another great advantage in the fact that dust can only lodge on the tops of the tubes (when the tubes are horizontal or sloping), leaving the lower surfaces, which are, of course, the most effective, always clean. The dust can be readily removed by means of steam-jets applied through suitable cleaning holes, provided in the brick-work sides of the boiler seating.

When water-tube boilers with vertical, or nearly vertical, tubes are used, dust cannot lodge at all, and as there is no soot in destructor gases, an externally clean heating surface is always presented. It may also be remarked that boilers of the water-tube type can be arranged closer to the end of the battery of destructor cells, and the damper arrangements can be made more convenient than when boilers of the Lancashire type are used.

35. *Fuel Economizers.*—Seeing that the destructor depends much more upon forced draught than upon the pull of the chimney, it is economical to carry the reduction of the temperature of the gases at the chimney base to a low point, say between 400 degrees and 500 degrees Fahr., and fuel economizers are frequently placed in the flues behind the boilers. These should be, like the boilers, arranged so that the gases can be by-passed at short notice.

36. *Utilization of Heat.*—The heat of the gases from destructors, although, as stated above, of considerable value, is unfortunately often lost through lack of convenient application near to the site of the destructor. It is a curious fact that, although electric-lighting stations only demand a considerable quantity of power during three or four hours per diem, the combination of destructors with electricity stations is the commonest method of utilizing the heat. There are, however, other kinds of municipal work that can absorb a fair proportion of the power available. Stone-breaking, crushing and screening the clinker from the destructors, or grinding it into mortar, driving repairing shops, sawing, chaff-cutting, sewage and water-pumping, heating baths and wash-houses, and even schools and dwellings, have been carried out in different places by means of the steam from destructors.

At Moss Side (Manchester) a snow melting pit has been constructed by the Municipal Engineer—Mr. Longley—in which a series of pipes bringing steam from the destructor boilers is placed. These are perforated so as to throw jets of hot steam upon the snow as it is tipped into the melting pit.

At Folkestone, the Borough Engineer—Mr. Nichols—has constructed heating pans on the flues beyond the economizer for the purpose of warming the materials for use in forming streets of tar Macadam.

37. *Dust-catcher.*—A dust-catcher has been placed between the boiler and the chimney in the installations at Brussels, West Hartlepool, Moss Side, and in many other recent destructors. The

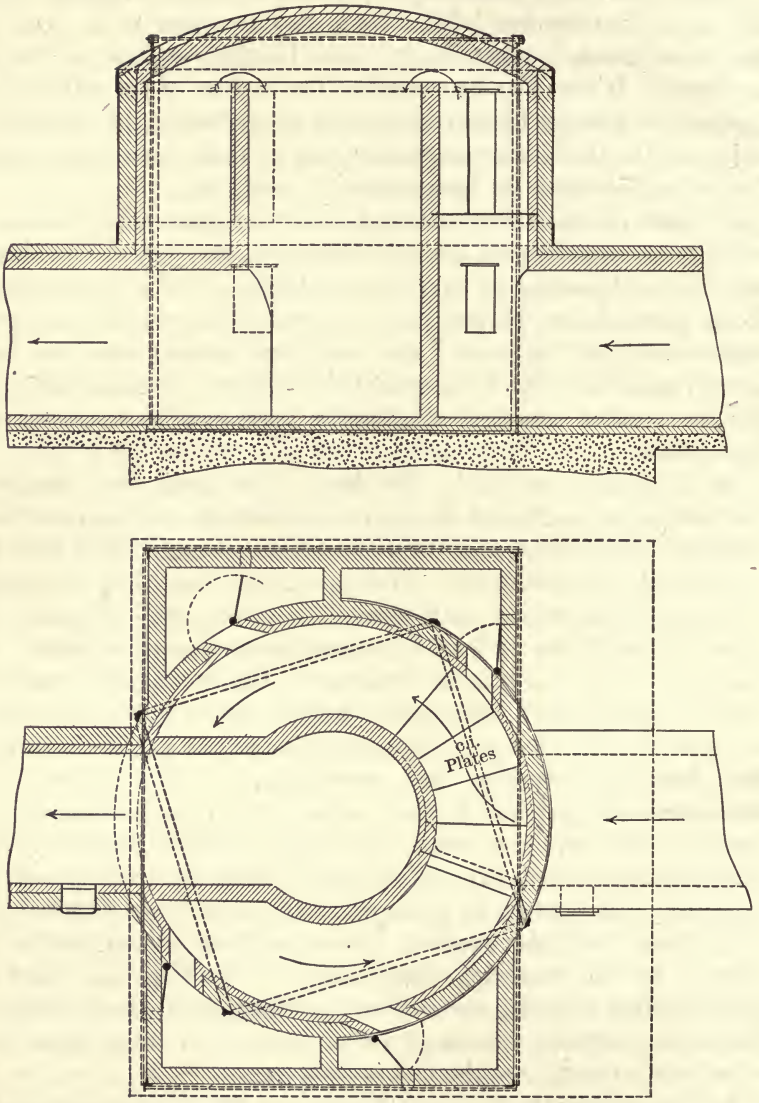


FIG. 523.—DUST-CATCHER.

5-Cell Destructor for Blackpool.

latest form of this apparatus has been devised by Messrs. Newton and Diggle, of Accrington, in conjunction with the author, and it consists of a swirling chamber, into which the gases are led from the main flue in such a manner as to give them a circular or revolving motion within the chamber, Fig. 523. In carrying this out they are made to pass round an annulus formed by the outer circular wall and the inner well, built up to within a short distance of the domed or arched roof. The whole of the interior is lined with fire-brick to enable the apparatus to deal with red-hot gases when the boilers are not in use. The rapid revolution of the gases within this chamber throws all the suspended dust—even the smallest particles—against the outside wall of the chamber by centrifugal force.

At intervals, vertical slits are provided, leading into pockets outside the chamber.

The dust, travelling along the outer wall, finds its way into these slits and thence into the pockets, where the gases are stagnant, and where the dust falls. The vertical slits leading into these pockets can be closed by suitable doors hung freely on ball-bearings and so arranged that whenever the outer cleaning door of the pocket is opened for the purpose of withdrawing the dust the door covering the inner opening is closed by suction.

With this apparatus it is possible to extract the dust from any or all of the pockets, without interfering with the draught of the destructor. This is important when destructors are combined with electricity stations, as at Accrington, because it is never practicable to allow the flues to get cool enough to clean out by hand. Not only the dust-catcher itself, but the whole of the flues throughout the destructor and boiler can be cleaned by simply opening the end doors, and allowing the wind to rush through, whipping up the whole of the dust deposited in the flues, and carrying it forward into the dust-catcher, where the centrifugal action discharges it into the pockets above described. It will be observed that the stronger the draught, the more complete is the separation of the dust.

38. *Dampers.*—Dampers of many kinds, including cast-iron (with and without ribs), wrought-iron and steel plates of different thicknesses have been tried. Cast-iron gas-valves, with cold-water circulation, have also been extensively used. The latter were successful until—as always happened sooner or later—the water supply was interrupted, by accident or design, when they would imme-

diately fail. Experience seems to point to a damper formed of fire-brick blocks in a suitably designed cast-iron frame as the most reliable for general purposes. The damper should be set in grooves deep enough to protect the frame from the action of hot gases. It should also be provided with suitable means for making the slit at the top practically air-tight, when the damper is either closed or open.

The author has recently adopted double by-pass dampers for the by-pass to the boilers. These are found to be practically gas-tight, as the damper nearest to the chimney so reduces the tension of the draught that the leakage past the second damper, with which the gases come earliest in contact, is negligible.

39. *Chimneys.*—A dissertation on destructor chimney shafts would be out of place in a paper which has already run to considerable length, but it may be remarked in passing that, unless in very special circumstances, a destructor chimney need not exceed from 100 feet to 120 feet in height. Even at the Westminster destructor, which is, as before stated, in the heart of London, the height of the chimney above ground line is only 90 feet, and the destructor is worked without complaint. In fact, it may be said that it is difficult to tell by observation of the chimney whether the plant is at work or not.

Destructor chimneys should be constructed to withstand the full heat of the gases, which are sometimes directed into them without passing through either boilers or economizers. For this reason they should invariably be lined to the top with fire-brick, and it is good practice also to provide an air space, properly ventilated, between the fire-brick lining and the outer shell.

40. *Clinker-handling.*—To come now to some of the machinery accessory to destructors, it may be mentioned that one of the most important points is the handling of the hot clinker as it is withdrawn from the furnace. At Moss Side, and also at Westminster, overhead railways, on the system devised by Mr. Cox, City Engineer of Bradford, and the late Mr. McTaggart, are used. These railways consist of a single H beam suitably suspended, on which a small trolley, with ball or roller bearings, runs freely, Fig. 519 and Fig. 524. From the trolley hangs a large tipping truck of suitable capacity for taking the whole of the clinker from the cleaning of a single cell. This represents two, and even sometimes three, very large barrow-loads, which would be heavy labor to wheel. The clinker truck is suspended immediately under the lip of the front

dead-plate, so that the clinker can be withdrawn directly into it. It is then pulled away by the stoker along the runway over the clinker cooling bed, and tipped on its own trunnions so as to discharge the clinker on the ground, where it is allowed to cool. These railways are capable of considerable modification to suit different sites. For instance, at Westminster, where the plant is sunk below ground level, a section of the rail is lifted on a

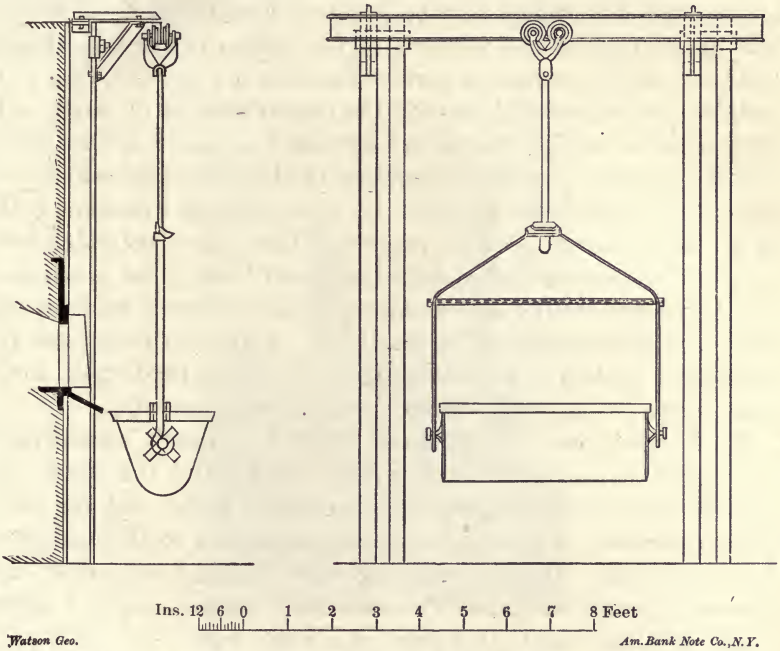


FIG. 524.—OVERHEAD CLINKER RAILWAY AND BUCKET. MOSS SIDE.

suitable hoist, and it is also provided with a revolving arrangement, so as to command several sets of rails at the top. Turntables are also easily contrived and worked.

41. *Clinker Crushers and Screens.*—Suitable clinker crushers and screens for preparing the material for concrete work, such as fire-proof floors, road foundations, and the like, have been designed also by Messrs. Cox and McTaggart, and are extensively in use. The crushers consist of a pair of fluted rollers mounted on wooden keys so as to give the necessary elasticity to avoid breakage when pieces of iron are encountered. The yield of the keys is sufficient to stop the machine gradually, and not instantaneously, and has the effect of throwing off the belt of the machine, and

avoiding the sudden jar which would cause breakage. The screens deliver the material in sizes required for the various works to be undertaken.

42. *Paving Flags.*—Attention has recently been given also to the production of concrete flags for footways, and to mixing and pressing machinery for making building-bricks.

43. *Clinker Bricks.*—Dr. Schultess, of Zürich, has worked out a very complete system for the production of bricks from clinker and lime. One of the features of his system is the lime-slaking machine, which produces a perfect lime in a dry powder, free from nodules, and slaked with exactly the proper amount of water. He also provides mixing machinery arranged so as to automatically mix the correct quantities of ground clinker, lime and water, and one of the advantages to which he draws special attention is the fact that no water need be expressed from the flag under compressure, the amount of moisture supplied being what is required for the proper setting and no more. This enables a brick or flag with very clean arrises to be produced. A further feature of Dr. Schultess's system is the maturing of the final product in forty-eight hours by means of steam at atmospheric pressure.

Bricks and flags so produced have a strength considerably exceeding that of good burned clay bricks, and the bricks are suitable for foundation work, embankment walls, and for inner walls of houses. Their dark color is a drawback to their adoption for outer walls. In the ordinary way Portland cement is used for such productions, but Dr. Schultess claims to get a better result with lime, and, of course, at a lower cost.

44. *Clinker Mortar.*—Mortar made from clinker and lime is found to be almost equal to hydraulic mortar, and it is capable of withstanding considerable heat. In fact, in destructors at Oldham, Bradford and elsewhere it is frequently used for pointing up the interior of the furnaces, and is found to stand as well as fire-clay.

45. *Destructors for Hospitals, etc.*—Destructors of small size, suitable for villages and public institutions, such as hospitals or large hotels, have been alluded to above. No special remark is called for respecting the design of such furnaces, except that they should be of the simplest character. As a rule they are both fired and clinkered at the front through the clinkering-door. In other respects, such as in the details of grate bars, cast-iron side-boxes, forced-draught apparatus, front-flue openings, and so forth, the

design of the larger types of furnaces is followed. In come cases, however, where it is impossible to get steam for steam-blast or electric current for fan draught, such furnaces have to be worked by means of strong natural draught.

46. *Portable Destructors.*—"Portable" destructors are also made, consisting of a furnace following the above general lines, attached to a plain cylindrical "multitubular" boiler with a dust-catcher in the smoke-box, steam-jet forced draught and a short chimney; the whole being mounted on wheels for transport. The

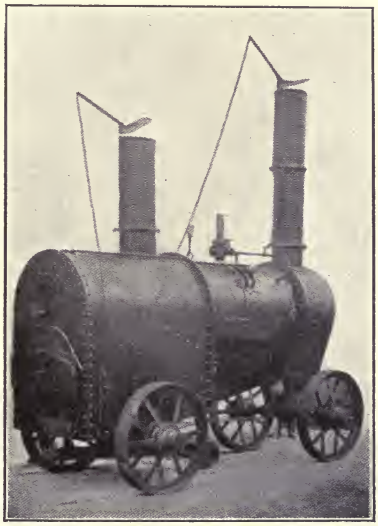


FIG. 525.—PORTABLE DESTRUCTOR.

weight of the destructor illustrated, Fig. 525, is between 6 and 7 tons, and the capacity about 4 tons of muck in the 24 hours. These small "portable" destructors are good steam-raisers, and when used in connection with hospitals the power may be conveniently utilized in connection with electric lighting plant, Röntgen ray apparatus, disinfecting chambers, and the like. They are also intended to be worked when stationary.

A still smaller destructor has been devised by the author's brother, Mr. F. L. Watson, for military purposes, consisting of a small furnace combined with sterilizing tanks, in which infected clothing, etc., can be dealt with, the whole being mounted on two wheels and suitable for mule or horse transport. There can be no doubt that a convenient apparatus of this kind, if available at all

standing and advanced camps, would materially lessen the ravages of zymotic diseases in time of war.

In conclusion, the author begs to acknowledge his indebtedness to many engineers, in not a few instances friends of his own, for their labors in connection with the disposal of refuse by fire, and he trusts that the above notes of his own experience may be of interest and of service to the members of the Institution.

APPENDIX I.

THE DESTRUCTOR AT WEST HARTLEPOOL.

Test Report.

Note.—The following figures and the accompanying diagram, Fig. 526, give the mean results of the two days.

Date of test	28th and 29th Jan., 1904.				
Duration of test	48 hours.				
Number and type of cells	6 back-to-back.				
Total grate surface	180 square feet.				
System of forced draught	Steam jets.				
Nature of refuse	Ashpit, nightsoil, market.				
Number of firemen and average wage per day	9 at 5 shillings.				
Number, size, and type of boilers	1 Babcock and Wilcox, 2,393 square feet.				
		Tons.	cwt.	qr.	lbs.
Total quantity of refuse burned	272,432 lbs. =	121	12	1	20
" " " per cell per 24 hours	22,703 lbs. =	10	2	2	23
" " " per square foot of grate per hour	31.5.				
Tons per man per shift	6.7 tons.				
Cost of labor per ton burned	8.9 pence.				
Total water evaporated	348,673 lbs.				
" " " per hour	7,264 lbs.				
" " " per square foot of heating surface per hour	3.03 lbs.				
" " " per lb. of refuse from and at 212 deg. Fahr. or 100 deg. C.	1.56 lbs.				
Mean steam pressure	155 lbs. per square inch.				
" feed temperature	43 degrees Fahrenheit.				
" main flue temperature	Above 2,000 deg. Fahr. (beyond range of Pyrometer).				
" temperature behind boiler	534 degrees Fahrenheit.				
Horse-power developed at 20 lbs. steam per indicated horse-power per hour	363.				
Purpose for which steam is utilized	Electric lighting.				

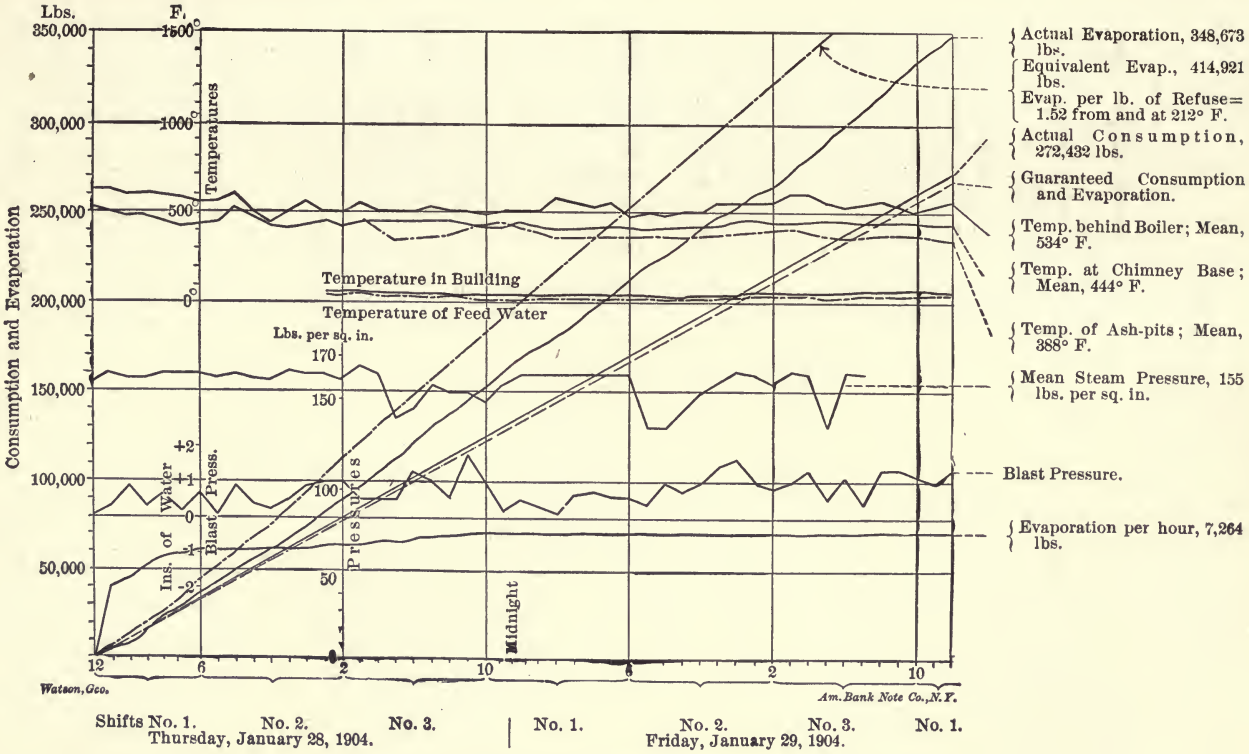


FIG. 526.—CHART OF TEST RESULTS, WEST HARTLEPOOL.
Hourly Observations. Temperature in Main Flue beyond Pyrometer reading.

APPENDIX II.

CITY OF WESTMINSTER.

Test Report.

Date of test	2d Dec., 1902, to 4th Dec., 1902.
Duration of test	45¼ hours.
Number and type of cells	6 back-to-back direct cart-fed.
Total grate surface	252 square feet.
System of forced draught	Steam jets.
Nature of refuse	House, trade, and market.
Number of firemen and average wage per day	9 stokers at 35 shil. per week. 4 top men at 27s. 6d. per week.
Number, size, and type of boilers	1 water-tube, 1,426 sq. ft. H. S.

	Tons.	cwt.	qrs.	lbs.	
Total quantity of refuse burned	138	15	1	0 =	310,828 lbs.
" " " " " per cell per 24 hrs.	12	5	1	8 =	27,476 lbs.
" " " " " per square feet of grate per hour	—	—	—	—	27.2 lbs.
Tons per man per shift	8	3	2	5	

Cost of labor per ton burned	11.5 pence, exclusive of engineman and foreman.
Total water evaporated " " " per hour " " " per square foot of heating surface per hour " " " per lb. of refuse from and at 212° F. or 100° C.	Test for evaporation abandoned, boiler blowing off very heavily.
Percentage of clinker and ash to refuse burned	24.9 per cent.
Mean steam pressure	125 lbs. per square inch.
" feed temperature	48 degrees Fahrenheit.
" main flue temperature	Well over 2,000 degrees Fahr. softened mild steel.
" temperature behind boilers	500 degrees Fahrenheit.
Purpose for which steam is utilized	Electric light, steam jets, hoist engines, etc.

DISCUSSION.

*Mr. J. Hartley Wicksteed.**—I know Mr. Watson very well, and I may say that he has been extremely successful with Mr. Horsfall's destructor. Destructors are very difficult things to undertake, because the town refuse varies so much in quality in the different towns, and it requires a very great deal of experience to hit off a result that you can guarantee. The original basis which differentiated Mr. Horsfall's destructor from others was the use of the steam jet. The action of the steam jet is very

* President of the Institution of Mechanical Engineers.

abstruse, but it has been reported upon by Lord Kelvin and Dr. Barr, and I will presently read to you one paragraph—not that I wish to direct your discussion to that particular point, but because it was what started this company making destructors. Of course, experience has shown that there are a hundred considerations which are of quite equal importance to the original idea concerning the utility of the steam jet, but still you must have something to start with, and Mr. Watson's company had this to start with and they worked at it and it lead them into the knowledge of all the other points. The thing itself I have, no doubt, is of considerable value. The author says in his paper, "it may be stated that they (that is, the steam jets) give a higher temperature and evaporation than the dry-air blast, provided that the refuse is rich enough in carbon to give the necessary temperature for the dissociation of the steam, which forms water-gas in the furnace, and which greatly improves the combustion.

This action has been explained by Lord Kelvin and Dr. Archibald Barr in the following words:

"A more important function is, however, fulfilled by the steam. In coming into contact with the incandescent fuel it is decomposed, the hydrogen being freed while the oxygen combines with the carbon in the fuel to form carbon monoxide. This decomposition of the water is effected by heat abstracted from the lower part of the fire, where it can be of comparatively small value for the cremation of the distillate. The water-gas (hydrogen and carbon monoxide) passes upwards to be burned by the excess air which it meets with over the fire, thus serving to increase the temperature which would otherwise exist at the meeting of the products of combustion with the gases distilled from the raw material."

So that it seems the chief function of the steam jet is to shift the locality of the intensest heat of the fire.

This paper and Mr. Russell's paper were prepared at the request of the American Society, and we from England have put before you the best knowledge possessed by any members of our Institution, and I hope you will take the matter up for discussion.

*Mr. Alfred Saxon.**—Sanitary engineering up to the present has had no particular attractions for me; in fact I have rather looked upon the science as a necessary evil, but seeing that the paper as presented seemed to my mind to be representative of the engineering of Leeds and its district, I thought I would see

71 * Member of the Institution of Mechanical Engineers.

whether no good thing could come from Manchester, which I represent, and, as a matter of fact, I believe we have the best destructor for the purpose produced in the Manchester district.

There are some points of agreement on this question, notably in respect to the quality of the refuse. I find that the people who have spoken and written upon this question agree with the statement of the author that an average for this is one-third by weight of water, one-third combustible matter, and one-third incombustible. On other points, of course, there are differences of opinion. The claim that Mr. Watson makes, for the isolated cell system at West Hartlepool works, does not, as a matter of fact, seem to be sustained, nor does it produce results which will compare with the continuous grate system which is adopted by the Manchester firm. The claim also made by the author about the ease of repairs in the isolated cells is doubtful, when one cell is adjacent to another, working at 2,000 degrees Fahrenheit, is it not rather too much to expect that much good work will be done in the cells which are supposed to be isolated, but which must to a certain extent be heated up by the cells which are in work.

Now, I should like to make a comparison of the test which is given in the paper with one which was made by the National Boiler Insurance Co. at Woolwich on the Destructor erected by Messrs. Meldrum Bros., Timperley. At Woolwich on three grates, 75 square feet, 63¼ tons of refuse were burned in 24 hours, and each pound of refuse evaporated 1.9 of water. Whereas at Hartlepool, in the test given in the paper, on 180 square feet, they burned 121 tons. In the one case, at Woolwich, at 77 pounds; whilst in the other case only 31 pounds per square foot.

Mr. Edward N. Trump.—While the papers presented by our English friends on the above subject are of great interest as showing engineering skill in handling a difficult problem, from the point of view of the chemical engineer the very name "destructor" suggests a process which should only be a last resort in the *utilization* of our wastes.

The remains of the beefsteak for which you paid two dollars a pound, and failed to eat because of a bad appetite; the butter left on your plate; and trimmings from the raw materials; which from such a large percentage of the food which is finally consumed, all contain valuable ingredients which should be utilized not destroyed.

The work in the United States has been more in the direction of the "reduction" of "garbage" or kitchen waste, to its useful

constituents rather than its destruction. The fermentation which so quickly starts up in these waste, producing a crop of germs very dangerous to the health of the community may be stopped, and the waste thoroughly sterilized by boiling under high pressure, and the valuable greases and liquids saved and utilized.

First, The waste must be gathered as frequently as possible to reduce the fermentation to a minimum. Second, It must be collected in covered carts or wagons, kept as air-tight as possible to prevent offensive odors escaping along the streets. Third, A number of large steel tanks or digestors of a strength sufficient to allow an internal steam pressure of one hundred pounds per square inch receive the garbage, forming an excellent storage for a fluctating delivery, and it is thoroughly cooked with direct steam until all of the dangerous germs are killed, the greases extracted and the whole thoroughly digested. Fourth, The products of this digestion are now utilized as follows: The contents of the digestors are emptied out in continuous filter, which separates the fibre from the liquid, pressing out as much of the latter as possible. The fibre is dried and may be burned in a special producer or furnace, with the production of large quantities of useful gas and of ammonia. The liquids are evaporated to get rid of the surplus water after the valuable greases are allowed to separate, and the resulting sirupy liquid mixed with the ashes from the fibre form valuable fertilizing. The greases are refined and often utilized in the production of toilets soaps. Many of the best known soap makers are using these greases from the reduction works.

Several kinds of machinery are already in use in this country to produce the above results. It only needs engineering skill to produce valuable products from the waste, and there is more than sufficient carbon in the dried fibre to make the heat and steam needed to operate the plant.

It would, therefore, appear that the *reduction* of waste rather than its destruction is the economical process to develop, and incineration should only be resorted to for those dry wastes which may be readily utilized as fuel, and which will stand storage without fermentation.

*Mr. George Watson.**—The author thanks the president for his very kind remarks. It is quite correct that Mr. Horsfall started his destructor business with the steam jet for forcing the draught. The steam jet, as pointed out somewhat briefly in the paper, has

* Author's closure, under the Rules.

several advantages, among which it may be again mentioned, that it protects the grate bars and other iron parts of the furnace from becoming red hot, while at the same time in many cases it increases the temperature for the combustion of the refuse. But it should not be supposed that the "Horsfall" system necessarily involves the use of steam jets.

In many cases it is found preferable to use fans, and care should always be taken to consider the exact conditions to be met in making the selection.

In a recent instance (at Guernsey) both steam jets and a fan have been provided in a "Horsfall" destructor.

This arrangement can now be made with little increase of cost over the fan system, and it is one which offers obvious advantages.

The remarks of Mr. A. Saxon serve to illustrate the difficulty in which the author was placed in preparing his paper, and which he endeavored to meet by confining his remarks strictly to his own experience. He also tried to avoid preferences as far as possible, and in indicating the advantages of the "Cellular" type of destructor, he expressed his own opinion, while endeavoring to avoid undue bias. Moreover, this is just one of the points constantly debated upon which the author considered himself free to speak his mind, for the reason that the "Cellular" system is no monopoly of any one firm, but is free to be adopted by all, as Mr. Saxon well knows. The author is not aware which of the claims put forward for the West Hartlepool Works is the one which Mr. Saxon considers is not sustained. He can only say that he is quite unaware of any inaccuracy in his statement. He must also say that in his opinion the results obtained at West Hartlepool are comparable to those obtained on either of the "Continuous-Grate" systems at present on the market, and he leaves members to make their own comparison; having regard to steam raising; not only on trial, but in every-day working; to cost of labor per ton burned; to durability; and to ease of repair. As to the expectation of being able to repair one cell while the others are at work which Mr. Saxon thinks unwarranted, the author can only say that under his own supervision such repairs are frequently made without difficulty.

While congratulating Mr. Saxon on the results obtained under test at Woolwich, the author may be allowed to say that there is no difficulty in burning a large amout of refuse per square foot

of grate per hour, either in the " Continuous-Grate " destructor, or in the " Cellular " destructor.

It is a question simply of strength of blast, and of frequency of raking over and of clinkering. In other words, the main consideration is that of the labor involved in the operation. It is the author's experience that much less labor is involved in burning at a rate of 30 pounds per square foot of grate per hour than at 70, while the clinker produced is harder, and the process is more complete, clinkering every two hours, than when clinkering at more frequent intervals.

In regard to the remarks of Mr. Trump, the author supposes it is more attractive to everybody to utilize than to destroy. But the engineer must divest himself of sentiment and reduce the question of reduction versus destruction to figures.

The impression gained in England as to the system of reduction prevailing in the United States is that, generally speaking, they do not pay, or at least that they will not continue to pay, and further it is practically impossible to conduct the process without creating a nuisance.

Such plants are often the subject of legal injunction, and where they escape the reason seems to be that they are placed at a considerable distance from dwellings.

In making these somewhat sweeping statements the author is painfully conscious of his lack of experience, but he is merely reflecting the general opinion which is held in Europe, which may be far from correct.

He does not understand how the digesters can form an equalizing storage for a fluctuating delivery, unless they can be discharged while under pressure, which he believes is not the case. Otherwise they must be completely charged and then sealed up till the process is complete, and in the meantime the refuse continually arriving must be otherwise stored. The author's own belief is that refuse, even in the United States, is constantly getting less valuable for any purpose, even for steam raising, as sanitary science progresses, and that if one-half of the skill employed to utilize the grease from the refuse were applied to the utilization of the clinker from the destructor, better results all round could be obtained from a centrally situated plant without nuisance, and with a large saving in cartage to put to the credit of the destructor.

No. 1251

THE HIGH-PRESSURE FIRE-SERVICE PUMPS OF MANHATTAN BOROUGH, CITY OF NEW YORK

DESCRIPTION OF PUMPS AND PUMPING SYSTEM WITH RESULTS OF TESTS

By Prof. R. C. Carpenter, Ithaca, N. Y.
Member of the Society

The object of this paper is to present a concise description of the high-pressure pumping system installed for fire service in the city of New York and the results of a test of the pumping machinery.

2 The system protects the district extending north from City Hall to Twenty-fifth Street, and east, approximately, from the North River to Second Avenue. It comprises about 55 miles of extra heavy cast-iron main, from 12-in. to 24-in. in diameter, with 8-in. hydrant branches; and two pumping stations so located that they never can be in the center of a conflagration. At the present time the pumping stations have a combined capacity of over 30,000 gal. per min. delivered at a pressure exceeding 300 lb. per sq. in.

THE SOURCE OF WATER SUPPLY

3 The supply of water is ordinarily obtained from the water mains of the city, which deliver Croton water to the stations at a pressure of from 14 lb. to 40 lb. per sq. in., depending upon the demand for water in that district. Both of the pumping stations are located close to tidal water and connections are made so that sea water can be obtained in case of difficulty with the Croton supply.

4 The advantage of the Croton water over salt water is that it is less likely to injure goods, and as the amount required for fire purposes is only a small percentage of that consumed for the daily supply of the city its use for fire protection makes no material difference from financial or insurance standpoints. As this is a matter of considerable importance data upon the quantity needed are given in the next paragraph.

Presented at monthly meetings, New York and St. Louis (October 1909), of The American Society of Mechanical Engineers.

WATER REQUIRED FOR FIRE PURPOSES

5 The general impression that an enormous quantity of water is required for fire purposes is erroneous as shown by figures furnished to Chief Engineer I. M. de Varona by the fire department for the Boroughs of Manhattan and Brooklyn, years 1900, 1901, 1902, 1903 and 1904. These give the average quantity of water used for fire protection during these years in the Borough of Manhattan as 74,010,803 gal. per year, of which 31,056,928 gal. was river water. The daily average use of Croton water, therefore, for the above five years was 117,000 gal.

6 For the Borough of Brooklyn the average for five years was 43,705,568 gal. of which 19,010,928 gal. was river water; daily average, 67,000 gal.

7 During these five years the greatest quantity used in the Borough of Manhattan was 99,000,000 gal. in 1901, which included 69,500,000 gal. of river water, leaving 29,500,000 gal. for Croton water, and Mr. de Varona states (Report of the Department of Water Supply, Gas and Electricity): "Even if this quantity be made 100,000,000 gal. per year, by comparing it with the average daily consumption of about 300,000,000 gal. it will be seen that the total amount used for fire purposes would be only about one-third of the amount used for all purposes in 24 hr., forming, therefore, an insignificant percentage of the total consumption. The quantity needed for fire purposes (one-tenth of one per cent) may therefore be entirely neglected as a factor in determining the water supply of the city.

8 "The capacity of each of the pumping stations will be for the present 15,000 gal. per min. or 43,000,000 gal. per day for the two stations. By the installation of three additional units in each station, for which provision is made, this capacity can be increased in round numbers to 69,000,000 gal. per day.

9 "The two stations, with the motors and pumps as installed, have a total capacity in excess of that of all the fire engines in the Boroughs of Manhattan, the Bronx and Brooklyn working under normal conditions. This comparison assumes the engines to work on one line of 2½-in. hose, say 500 ft. long, under a pressure of, say 200 lb., and with the capacities as printed in the official blank forms of the reports of the fire department. It should furthermore be remembered that provision is made for the installation of still another pumping station."

MOTIVE POWER

10 The power for driving the pumps is transmitted electrically from several of the electric power and lighting systems located on Manhattan Island. As the stations of systems are widely separated and any or all of them are available for motive power the system of electric transmission was considered more reliable in the case of a large and general conflagration than power plants maintained directly at the pumping stations. Each station is provided with two independent sets of transmission lines located as far as possible beyond danger or injury in case of a great conflagration.

11 The cost of erecting and maintaining an independent power plant would have entailed a greater annual charge than the cost of the electric current; consequently the present arrangement is advantageous from a financial standpoint.

12 In addition to the charge per kilowatt for the current delivered there is a charge aggregating $90,000 per year for reserving the first right of use for the necessary generating machinery for this purpose. The total cost of maintenance of the system is estimated at $170,000 a year, which amount it is believed will be saved many times over by a reduction in insurance premiums now paid in the protected district.

13 The electric current is supplied at a pressure of 6600 volts from the following stations of the New York Edison Company, having the capacity indicated: 53 Duane Street, 7600 kw.; 115 East 12th Street, 1700 kw.; 45 West 26th Street, 400 kw.; 140th Street and Ryder Ave., 4000 kw.; Waterside Stations No. 1 and No. 2, 196,700 kw. In addition there are feeders extending to the Brooklyn Edison Company stations which can be called on in case of an emergency demand.

14 The pumping stations are connected to 18 sub-stations, equipped with rotary converters and storage batteries, aggregating a capacity of 124,000 ampere hours at 135 volts, an enormous reserve.

15 Each station is connected with the main stations of the Edison Company by two 250,000 c.m. three-phase cables laid in ducts, and two independent reserve feeders extend to the sub-station system of the Edison Company. With all these precautions, interruption of the power supply would seem a physical impossibility.

THE DISTRIBUTION SYSTEM

16 The following information upon the distribution system is taken largely from the department report of Chief Engineer de Varona

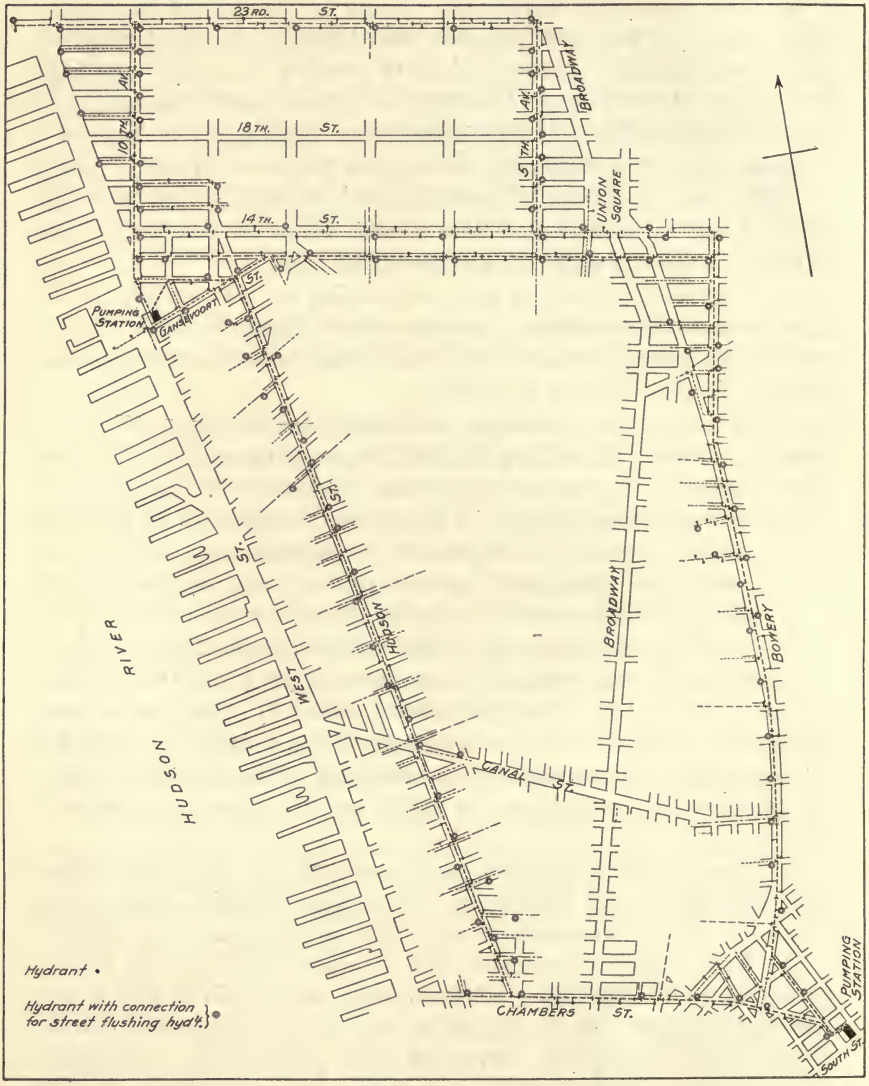


FIG. 1 SHOWING LOCATION OF STATIONS AND AREAS COVERED BY HIGH-PRESSURE PUMPING SYSTEM

THE AREA INDICATED IS SERVED BY A SYSTEM OF MAINS RANGING FROM 24 IN. TO 12 IN. IN DIAMETER WITH 8-IN. HYDRANT CONNECTIONS

for 1905. Fig. 1 shows the system to be bounded by mains laid on the north through Twenty-third Street; on the east, through Broadway to Fourteenth Street, through Fourteenth Street to Third Avenue, down Third Avenue to the Bowery, down the Bowery to Chambers Street; through Chambers Street on the south to West Street; and on the west through West Street.

17 The area actually protected is considerably greater than this as hose can be extended over a zone 600 ft. wide beyond the limits of the mains.

18 This district was selected as that in which the fire losses were the greatest and which most urgently needed fire protection. Plans have been prepared for the extension of the system southerly to the Battery, easterly as far as the East River, and, if necessary, northerly as far as Fifty-ninth Street, by the simple extension of the mains and probably the erection of a third pumping station.

19 The pipes, castings and hydrants were tested at a pressure of 450 lb. The specified allowance for leakage in a 10-min. test was at the rate of 4 gal. in 24 hr. for each lineal foot of pipe joint, equivalent to a leakage of 487,000 gal. for the whole system in 24 hr., which is somewhat over one per cent of the total specified pumping capacity now installed. The actual leakage on test was at the rate of 264,000 gal. per day or about six-tenths of one per cent of the pumping capacity. Considering the difficulties of construction and the high pressure, the results attained were remarkable and reflect great credit on the engineer in charge.

20 There are sufficient hydrants so that if a block were on fire 60 streams of 500 gal. per min. each, or the full capacity of both stations, could be concentrated on a block with a length of hose not exceeding 400 ft. to 500 ft., assuming the use of 3-in. hose and $1\frac{1}{4}$-in. nozzles.

21 The layout of the mains at the stations both for suction and delivery is on the loop system; that is, the supply can be taken from either one of two mains, and the discharge is through either one or both of two mains. With this system even the breakdown of one of the discharge mains at the station would only slightly reduce the pressure at the fire and would not affect the capacity of the station, as the pumps would be capable of forcing their full discharge through the short length of a single 24-in. main if made necessary by such an accident.

22 The mains are of cast-iron, bell and spigot pipe, of the thicknesses given in the following table:

Size of Pipe Inches	Thickness Inches	Unit Tensile Strain with 300 lb. pressure	Factor of Safety
24	1⅞	1920	0.4
20	1½	2000	10.0
16	1¼	1920	10.4
12	1	1800	11.1
8*	⅞	1371	14.6

* Used only for hydrant branches.

SUPPLY PIPING

23 At the South Street Station the fresh water supply is derived from two 30-in. lines, one connected at Chestnut Street to the 36-in. line on Madison Street, and the other connected at Pike Street to the 36-in. line on Division Street. These two main feeders, to which the two 30-in. lines are connected, increase to 48 in. in diameter and extend independently and directly to the Central Park Reservoir and are also reinforced by connections with the main feeders in this section of the city.

24 An auxiliary salt-water supply, consisting of two 36-in. pipes about 140 ft. long, brings the salt water from the East River to a suction chamber located directly in front of the pumping station. This suction is so constructed that the pipes are always below mean low water, thus insuring a supply at all times and avoiding the possibility of a break in the suction caused by air getting into the suction lines. On the river end of this suction there are constructed heavy bulkhead screens and in the suction chamber are two sets of bronze screens which are readily accessible for cleaning. From the suction chamber there are taken two 30-in. flanged mains to the duplicate set of mains in the pumping station proper. The vacuum in these 30-in. pipes is maintained by automatic electric vacuum pumps located on the pump room floor of the station.

25 At the Gansevoort Street Station the fresh-water supply is derived from two 30-in. mains, one connected at Twelfth Street to the 48-in. line on Fifth Avenue, which runs direct to Central Park Reservoir, and the other connected to the 36-in. line on Ninth Avenue at Little West 12th Street, which increases to a 48-in. line and runs also direct to the Central Park Reservoir. These two main feeders, in addition to having their supplies direct from Central Park Reservoir, are also reinforced by connections with the main feeders in this section of the city.

26 The salt-water suction lines for this station are practically identical with those for the South Street Station except that the 36-in. lines from the North River to the station are 650 ft. long.

PUMPING STATIONS

27 The two stations, known as the Gansevoort pumping station, located near Gansevoort Market on the North River, and the South Street station, located on the corner of Oliver and South Streets near the East River, are identical in construction and equipment. The buildings are of simple design, of steel fire-proof construction, with concrete foundations. The Gansevoort Street building, which is typical of both, is one story high with basement, 63 ft. 8 in. by 97 ft. 4 in. Each station is large enough for eight pumping units.

MACHINERY

28 There are now five units in each station consisting of Allis-Chalmers five-stage centrifugal pumps driven by Allis-Chalmers induction motors and the necessary auxiliary machinery. The motors and pumps are alike and their parts are interchangeable.

29 The pumps each have a specified capacity of 3000 gal. per min. of sea-water, working with a suction lift of 20 ft. and a delivery pressure of 300 lb. per sq. in. The actual capacity as indicated by a 24-hr. test was about 30 per cent in excess of that specified. The original specifications contemplated the use of six-stage pumps, with the expectation that sea-water would be used at each fire. Because of the facts already referred to (Par. 4), that the relative amount of water required for fire purposes is insignificant and that sea-water may do considerably more damage to goods than fresh water, a change in the specifications was agreed to, whereby the pumps should work at best efficiency when receiving water from the Croton mains at a pressure on the intake side varying from 15 lb. to 40 lb. per sq. in.

30 To meet this new condition the pumps were all built with five stages. All the sea connections and priming machinery as originally contemplated were installed, so that sea-water can be pumped into the mains whenever desired. The effect of the change is merely to reduce the pressure head slightly in case sea-water is used.

ARRANGEMENT OF MACHINERY

31 The floor-plans of the buildings and general layout of machinery, piping, switchboards, etc., are shown in Fig 2. As will be seen

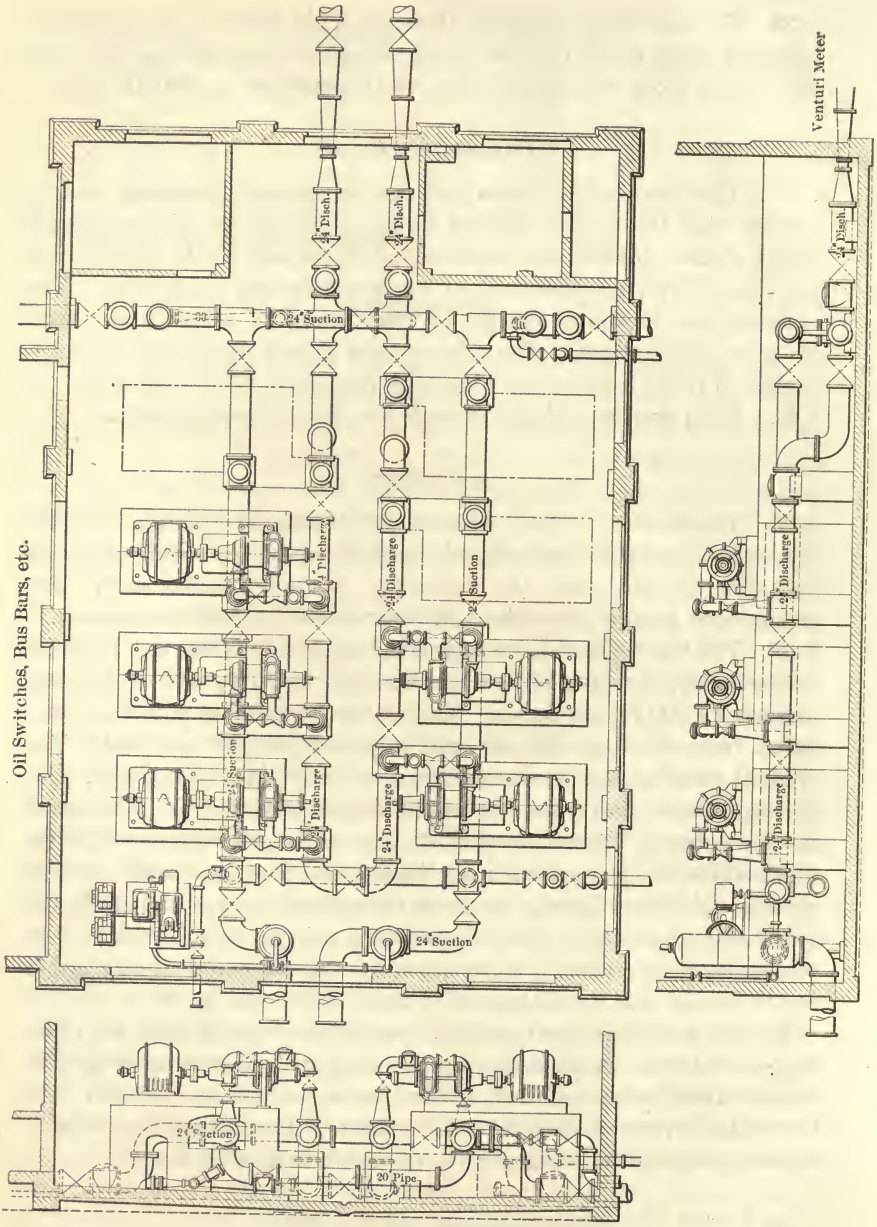


FIG. 2 PLAN AND ELEVATION SHOWING ARRANGEMENT OF HYDRAULIC AND ELECTRICAL APPARATUS IN PUMPING STATIONS

space is provided for three additional units. Working detail plans of the machinery were furnished by the contractor. The arrangement shown in Fig. 2 is the same for both stations, the only difference being that the switchboard and office in the South Street station are on different sides of the building as compared with the Gansevoort Street station.

32 The motors and pumps, with suction and delivery branches, are located on the main floor of the pump room. The switchboard and switchboard apparatus are placed in an enclosed two-story and basement gallery.

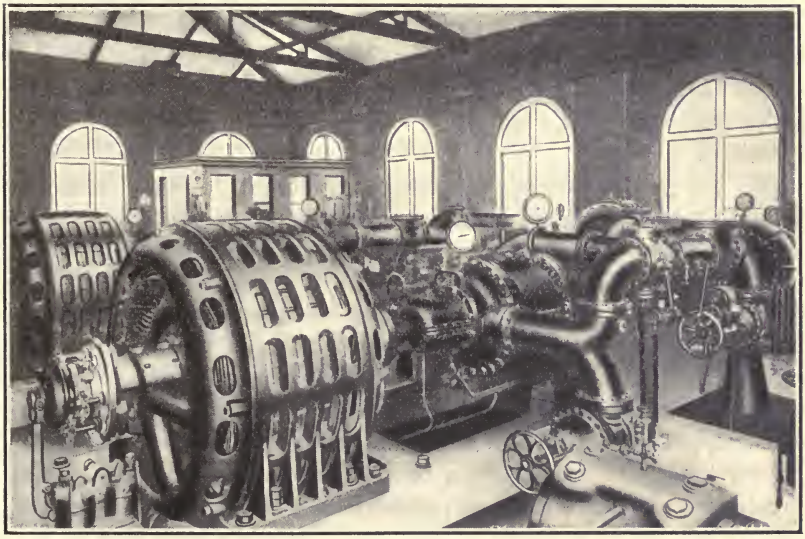


FIG. 3 INTERIOR VIEW OF STATION

33 The four high-tension feeders and all other wires entering the building are brought in through the gallery basement. All terminal work on the entering wires is located in the basement. On the first floor of the gallery, which is approximately on the same level as the pump-room floor, are placed the oil switches, with their controlling and protective devices, fire-proof cells and compartments.

34 The operating switchboard is conveniently located in the enclosing wall of the gallery, and is so placed as to allow a man standing on the pump-room floor to perform all the operations necessary for controlling the apparatus in the station. The bus bars,

with their fireproof compartments, are placed on the second floor of the gallery.

MOTORS FOR CENTRIFUGAL PUMPS

35 The motors are of the constant-speed, wound-rotor induction type, 3-phase, 25-cycle, 6300-volt to 6600-volt, designed to operate at about 740 r.p.m. Each pump is direct-connected to its motor by a flexible coupling which takes care of any variation from alignment. In starting, an iron grid resistance is connected in the secondary circuit and gradually cut out by means of a handwheel on the motor switchboard panel. When the resistance is all cut out the rotor is automatically short-circuited and operated by specially constructed solenoids through a small switch mounted directly on the shaft of the handwheel above referred to. An interlocking arrangement prevents the operator from closing the switch connecting the motor to the line while the motor is short-circuited.

36 The specifications required the motors to have sufficient starting torque to attain full speed between 30 sec. and 45 sec. after starting, with a current not exceeding 150 per cent of that used when the motor is working under full speed. Each motor was required to develop not less than 800 b.h.p. when using current of 6300 volts, 25 cycles, and under these conditions to have an efficiency not less than 92 per cent, a power factor not less than 93 per cent, and a motor slip not in excess of 2 per cent. At three-quarters load the efficiency was not to be less than 92 per cent and the slip not to exceed 1.5 per cent. It was specified that the temperature of the motors should not rise more than 40 per cent on a 24-hr. test at full load, when measured by a thermometer, the air in the room being 25 deg. cent.

37 Prof. Geo. F. Sever of Columbia University tested two of the motors in the shops of the contractor and found them to meet the specifications and to have a full-load efficiency of 93.2 per cent. The other motors were inspected and found to be alike and were assumed to have the same efficiency. The motors were also tested for temperature rise at the time of the official test to be described later.

MOTORS FOR AUXILIARIES

38 Direct-current motors of 240 volts are provided to operate the various gate valves in the station and the piston pumps employed for maintaining a vacuum on the salt-water suction lines.

PUMPS

39 As previously stated the pumps were finally constructed with five stages, each to give a pressure of somewhat over 60 lb. per sq. in., making the combined pressure of the five stages about 300 lb. per sq. in. above the intake pressure, which is the maximum working pressure of the stations at normal speed of 740 r.p.m. This type of pump is the simplest now on the market for pumping water either against a high head or low head, and this simplicity was the deciding factor which led to the selection of this style of machinery.

40 The pumps are water-balanced by a piston connected to the last impeller and upon which the water pressure acts, but should any additional end-thrust occur, it would be taken up by the ball bearing provided in the outboard bearing. This ball bearing consists of two rings of 1½-in. diameter steel balls and is water-cooled. The balancing piston is fitted very loosely in order to keep the friction losses small, and as a result a considerable amount of water leaks past it into a chamber at the end of the pump, which is provided with a discharge pipe and valve leading into the suction. By adjusting the valve in this pipe the difference of pressures on the piston can be regulated as desired. The bearings are of the ring-oiled type and are separated from the pump casing by packing glands which prevent foreign matter from entering the bearings. The impellers are of bronze and the shaft of forged steel. All parts of the runners and diffusion vanes are thoroughly lubricated by oil cups on the base of the pumps. A feature is the wide base, shown in Fig. 4, which allows the pump barrel to set low, giving stability.

41 Each combined unit is equipped with automatic and hand control. The pumps are kept primed for instant service and the simple operation of a switch on the main switchboard starts the machine and gives full pressure in about 30 sec.

PRESSURE-REGULATING VALVES

42 A combined regulating and relief valve is interposed between the discharge pipe and the suction pipe of each pump, and set to regulate the discharge of each pump to any predetermined pressure.

43. When the volume of the water discharged by the pump is in excess of that forced into the system, this valve acts as a relief valve and by-passes this excess into the suction to the pump, the pressure on the main distribution system remaining at the predetermined

point. When no water is forced into the distribution system all of the water discharged from the pump is then by-passed into the suction.

44 The pressure-regulating valves were made by the Ross Valve Mfg. Co., of Troy, N. Y., and much of the practical success of the station has been due to the accuracy with which they maintain any desired pressure.

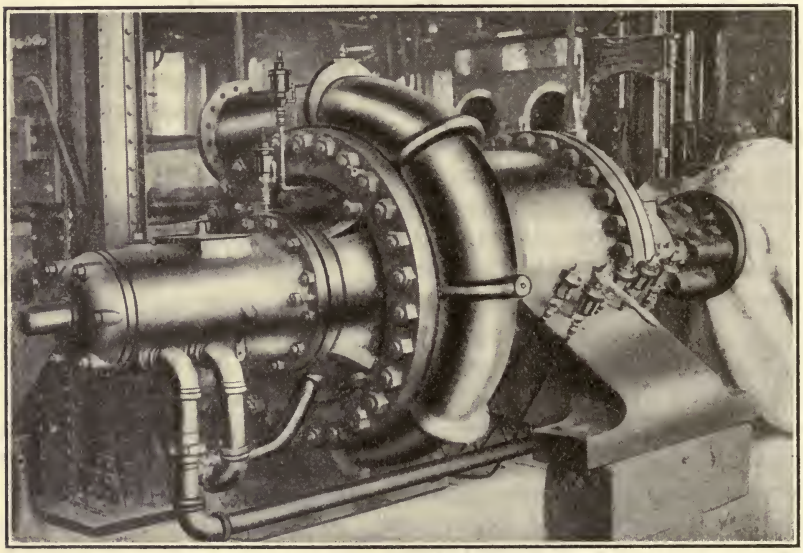


FIG. 4 MULTISTAGE PUMP, CAPACITY 3000 GAL. PER MIN.; MAXIMUM HEAD, 300 LB. PER SQ. IN.

PRIMING APPARATUS FOR SALT-WATER SUCTION LINES

45 The priming apparatus in each station consists of three motor-driven vacuum pumps, each arranged to maintain automatically a vacuum of 26 in. in the suction lines. These pumps are of the piston single-action type, one having a displacement capacity of 300 cu. ft. per min. for a piston speed of 200 ft. per min. and each of the others a displacement capacity of 50 cu. ft. with a piston speed of 160 ft. per min.

46 An air-collecting chamber is connected to each of the salt-water suction lines and equipped with water-gage glass and vacuum gage. The air-suction piping between the air chambers and the air pumps is provided with a vertical loop sufficiently high to prevent

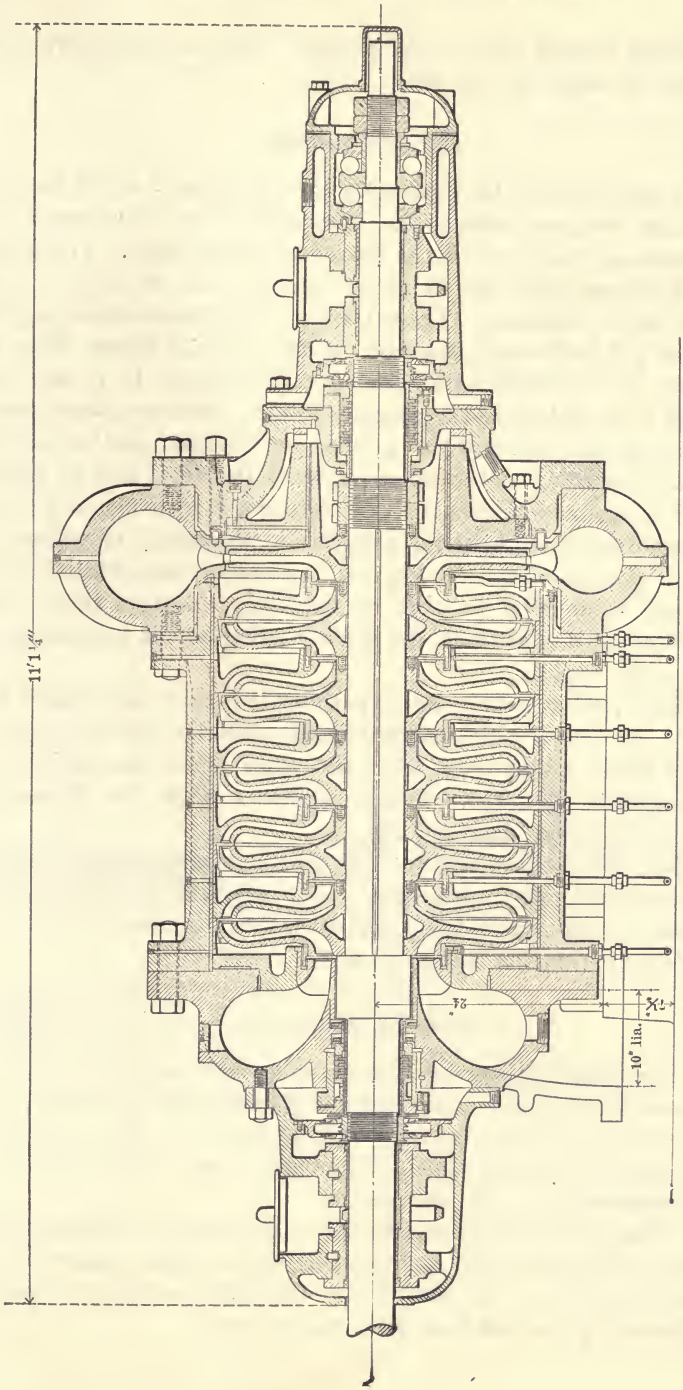


FIG. 5 CROSS-SECTION OF FIVE-STAGE, HIGH-LIFT CENTRIFUGAL PUMP

water being carried over to the pumps. The air pumps are interconnected to each air chamber.

VENTURI METERS

47 Venturi meters for measuring the discharge of water from the station and from one main to the other were set by the contractor on each discharge main and on the cross-connecting main. The meters of the discharge main are 24 in. in diameter and on the cross-over main 12 in. in diameter. These meters were standardized under the direction of F. N. Connet, Manager of the Venturi Meter Sales Department of the Builders Iron Foundry, Providence, R. I., and were provided with dial-indicating gages and also chart-recorders graduated to indicate the flow in gallons per minute; and in addition with an integrating meter which registers the total flow in gallons.

48 The readings during the test were taken by a mercury manometer, graduated to show the capacity in thousands of gallons per minute. For this purpose a Venturi manometer was attached with a temporary connection to each of the 24-in. Venturi tubes. The manometer gave essentially the same reading as the indicating dial on the main register.

49 The Venturi manometer is practically a tube partly filled with mercury, one side of which communicates with the upstream pressure chamber of the meter tube, while the other communicates with the throat-pressure chamber. The connections with the manometer are indicated in the diagram, Fig. 6.

50 The sketch shows a 24-in. high-pressure meter tube, its register-indicator-recorder and manometer. The instruments and meter tube are drawn to scale, but in the pumping station the meter tube is about 75 ft. distant from the instruments.

TESTS OF MACHINERY

51 The specifications for the pumping system provided for an endurance test of each motor and pump lasting 24 hr. without stop, during which time the capacity and efficiency of the pumps and motors were to be determined. The tests were to be in charge of an expert appointed by the commission.

52 The specifications provided for making the test with sea water, but this was later changed to a test with Croton water under the conditions of actual use. In view of this change the contractor increased the efficiency guarantee from 70 to 71 per cent.

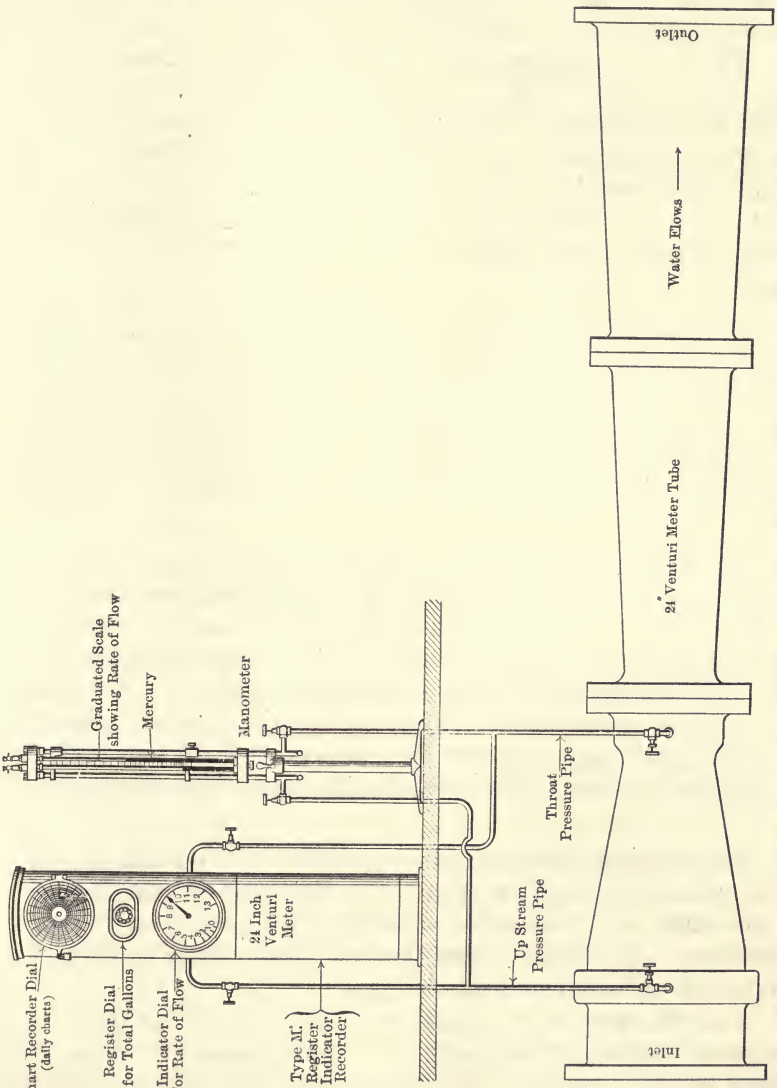


FIG. 6 ARRANGEMENT OF MANOMETER AND ITS CONNECTIONS TO THE VENTURI METER

53 The original specifications called for a capacity of 3000 gal. of sea water per minute against a discharge pressure of 300 lb. per sq. in. and a suction lift not exceeding 20 ft. The total increment of pressure is equivalent to 308.66 lb. from the intake to the delivery side. The Croton pressure varies at the stations in different parts of the day from about 40 lb. to 13 lb. per sq. in. and is affected by the amount of water being drawn from the mains. Consequently, to meet the requirements, the delivery pressure would need to be 308.66 lb. in excess of the intake pressure. There is also a further correction from the fact that sea water is heavier than fresh water and this correction under maximum conditions might amount to 2.5 per cent.

54 The specifications further provided that the brake horse-power developed by the motors under test should be computed from the electrical energy supplied to them, corrected for the efficiency of the motors as determined by the test. They further provided that if the aggregate of all stops exceeded one hour for any motor the test for capacity for such motor was to be run over again for a period of 24 hr.

55 The specifications also provided that the pumping capacity of the apparatus and the efficiency of the pumps should be based on the minimum rate of pumping during any eight consecutive hours of the endurance test, during which none of the motors were stopped.

56 The discharge of the pumps was determined by the reading of the Venturi meters, one of which was located in each discharge line. These readings were under the direction of F. N. Connet, and were checked by observers representing the contractors and also the city.

57 The modified specifications also required that the efficiency of *each pump* should be not less than 70 per cent and its capacity not less than 3000 gal. of sea water when lifted to a pressure equivalent to 308.66 lb. To determine whether the requirement was met, a separate test of each pump was required.

58 The efficiency of the pumps was computed by dividing the horse-power output of the pumps by the horse-power input as received from the motors. The horse-power input was computed as follows:

$$\text{h.p. input} = \frac{\text{total watts}}{746} \times \text{efficiency of motors (93.2 per cent)}$$

The horse-power output was computed as follows: h.p. output $= \frac{\text{wt. per gal. (8.34)} \times 2.31 \text{ head in pounds} \times \text{no. of gal. per min.}}{33{,}000}$

SOUTH STREET STATION TEST

59 The test of the South Street Station was begun at 12:30 p.m. on September 2, 1908, after about 2 hr. of preliminary running for the purpose of adjusting the delivery pressure; it was continued without interruption for 24 hr. With the exception of a short stop of motor No. 2 which was shut down from 2:11 to 2:41 a.m., September 3, to remedy a slight defect in the insulation of the field coils, no pump was stopped. During the time No. 2 was stopped the pressure on the delivery mains fell to about 300 lb.; during the remainder of the test the pressure was maintained at or above the contract requirement, as will be noted by consulting the last column of Table 1.

60 The average results for each hour for the 24-hr. test of all four motors are given in Table 2. The smallest delivery for eight consecutive hours occurred at the last part of the test, when the average capacity, as shown by the readings, was 18,447 gal. per min., and the average efficiency was 72.2 per cent. During this time the average pressure pumped against was 314.5 lb., or an excess of about 6 lb. over contract requirement.

61 It will be noted from the last column of Table 2 that there is considerable variation in the efficiency; that during the first hour the efficiency was less than 70 per cent, whereas during the third and fourth hours the efficiency exceeded 75 per cent. This variation in efficiency was doubtless caused by variation in the amount of water by-passed from the pressure to the suction side of the pump over the balanced piston and through the bearings, and possibly during the first hour by the discharge of some water through the relief valve which was pumped but not metered. The valves for regulating the differential pressure on the balance pistons were nearly closed during the third, fourth and fifth hours of the South Street Station test, but were opened the normal amount for the remaining portion of the test. The amount of water which for maximum difference of pressure may leak around the balance piston of any pump without passing through the meter could not be accurately determined but was estimated to be in excess of 4 per cent. Hence it appears that slight changes in the opening of the valve controlling the differences of pressure at this piston must materially affect the efficiency. The normal

opening of this valve appears to correspond to an efficiency of about 72.5 per cent.

62 During the test of the South Street Station all the bearings ran cool with the exception of those on No. 6 pump, which heated up during the third and fourth hours but were brought to a normal condition without stopping the pump or reducing its load by the application of lubricants and cooling water.

TABLE 1 HOURLY AVERAGE OF READINGS OF DISCHARGE AND INJECTION GAGES ON PUMPS

SOUTH STREET PUMPING STATION, SEPTEMBER 2 AND 3, 1908

Hour	Pump No. 6		Pump No. 4		Pump No. 2		Pump No. 1		Pump No. 3		Average		Net Pressure Lb.
	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	
12:30– 1:15	335.0	21.8	332.3	22.6	329.8	22.5	331.0	24.0	332.9	23.3	332.2	22.8	309.4
1:30– 2:15	347.2	20.9	347.0	22.2	345.5	22.1	343.8	22.6	346.2	22.1	345.9	22.0	323.0
2:30– 3:15	345.0	20.9	343.8	21.8	342.8	22.1	341.0	22.5	343.4	21.8	343.2	21.8	321.4
3:30– 4:00	344.2	20.9	343.8	21.9	340.3	22.3	339.8	22.6	342.2	21.9	342.1	21.9	320.2
4:30– 5:00	341.7	21.1	341.8	21.9	340.8	22.6	339.3	23.1	342.2	22.4	341.2	22.2	319.0
5:30– 6:00	336.2	22.4	336.3	22.9	335.3	23.8	333.3	25.3	335.7	23.6	335.4	23.6	311.8
6:30– 7:00	337.2	24.4	336.8	24.9	334.8	25.3	333.8	26.3	335.7	24.9	335.7	25.2	310.5
7:30– 8:00	339.7	25.1	337.8	25.9	335.8	26.6	334.8	27.3	337.7	25.9	337.2	26.2	311.0
8:30– 9:00	341.2	26.6	339.3	26.9	338.3	27.3	337.3	28.8	338.7	27.1	339.0	27.3	311.7
9:30–10:00	344.7	27.6	342.8	27.9	341.8	28.6	343.3	29.3	344.7	27.9	343.5	28.3	315.2
10:30–11:00	342.2	28.9	341.3	29.1	340.3	29.3	340.8	30.8	342.7	29.6	341.5	29.5	312.0
11:30–12:00	343.7	30.4	344.3	29.9	343.3	30.3	342.3	32.1	343.7	30.6	343.5	30.7	312.8
12:30– 1:00	345.7	30.6	347.3	30.1	347.3	30.8	345.8	32.6	347.9	31.4	346.8	31.1	315.7
1:30– 2:15	334.0	30.9	334.5	30.6	*		332.6	33.1	333.9	31.4	333.7	31.5	302.2
2:30– 3:00	332.4	31.1	331.0	30.9			330.8	32.8	332.2	31.6	331.6	31.6	300.0
3:30– 4:00	349.2	31.4	348.3	31.1	346.3	31.6	347.3	33.3	349.2	31.6	348.1	31.8	316.3
4:30– 5:00	347.2	31.1	346.8	30.9	346.3	31.3	345.8	32.8	349.4	31.4	347.1	31.5	315.6
5:30– 6:00	346.7	28.6	346.3	28.4	345.3	29.3	345.3	30.6	348.2	29.4	346.4	29.3	317.1
6:30– 7:00	342.2	27.6	342.3	25.4	340.3	26.1	340.3	27.1	341.2	26.1	341.3	26 5	314.8
7:30– 8:00	332.7	21.6	332.3	22.1	330.3	22.6	329.3	24.1	331.2	22.6	331.2	22.6	308.6
8:30– 9:00	332.7	20.9	332.3	21.4	331.3	22.1	329.3	22.6	331.2	21.6	331.4	21.7	309.7
9:30–10:00	331.7	20.6	332.3	20.9	328.8	21.8	328.3	22.8	331.2	22.1	330.5	21.6	308.9
10:30–11:00	334.2	21.4	336.8	21.6	332.8	22.3	331.8	23.3	336.2	22.4	334.4	22.2	312.2
11:30–12:30	336.0	21.9	338.0	21.9	333.6	22.6	334.0	23.8	337.7	22.6	335.9	22.6	313.3

Readings corrected for error of gage and to center of pumps.
* Pump No. 2 shut down from 2:11 to 2:41 on account of motor.

63 It will be noted from Table 2 that the average results of the 24-hr. test of the South Street Station exceeded the contract requirements in capacity, pressure head and efficiency.

64 The horsepower delivered by the motors during the test averaged for the 24 hr. about 920 or about 15 per cent above rating, without excessive heating.

TABLE 2 COMPUTATION OF PUMP EFFICIENCIES

SOUTH STREET PUMPING STATION, SEPTEMBER 2 AND 3, 1908

Hour Beginning	r.p.m.	Total kw. per hr.	Total h.p. from motors (input)	Gal. per min.	Net pressure lb.	h.p. delivered	Efficiency per cent
12:30 p.m.		3875	4841.4	18334	309.4	3311.6	68.6
1:30	757.0	3851	4811.4	18634	323.9	3523.6	73.2
2:30	755.0	3829	4784.0	19217	321.4	3605.7	75.4
3:30		3819	4771.5	19220	320.2	3592.8	75.3
4:30	755.0	3811	4761.5	19145	319.0	3565.4	75.0
5:30		3837	4794.0	18995	311.8	3457.6	72.1
6:30	755.0	3818	4770.3	18970	310.5	3438.7	72.2
7:30	755.0	3815	4767.5	18980	311.0	3446.0	72.3
8:30	757.0	3863	4826.4	19020	311.7	3461.1	71.6
9:30	756.0	3868	4830.2	19120	315.2	3518.3	72.8
10:30	757.0	3873	4838.9	19095	312.0	3478.1	72.1
11:30	756.5	3859	4821.4	19120	312.8	3491.6	72.4
12:30 a.m.	757.0	3870	4835.2	19175	315.7	3534.1	73.0
1:30	756.5	3672	4587.8	18790	302.0	3315.0	72.5
2:30	757.0	3667	4581.6	18776	300.0	3288.4	71.5
3:30	757.0	3890	4860.2	19190	316.3	3543.5	72.8
4:30	757.5	3891	4861.4	19160	315.6	3530.2	72.6
5:30	754.7	3861	4823.9	19110	317.1	3337.7	73.2
6:30	757.0	3865	4828.9	19005	314.8	3492.7	72.4
7:30	754.7	3706	4630.4	18710	308.6	3370.8	73.0
8:30	745.6	3659	4571.5	18100	309.7	3272.5	71.5
9:30	745.6	3651	4536.5	17890	308.9	3226.2	71.5
10:30	745.0	3619	4521.6	17795	312.2	3243.4	71.8
11:30	747.0	3618	4519.1	17806	313.3	3256.8	71.8
Average	756.1						72.5

Average efficiency, 1st period of 8 hr. = 73.0 per cent.
Average efficiency 2nd period of 8 hr. = 72.3 per cent.
Average efficiency 3rd period of 8 hr. = 72.5 per cent.

No. of cycles per sec. 12:30 p.m. to 6:30 a.m. = 25.5
No. of cycles per sec. 6:30 a.m. to 12:30 p.m. = 25.0

TABLE 3 TEST OF INDIVIDUAL PUMPS

SOUTH STREET STATION, SEPTEMBER 3, 1908

Time	No. of pump	Gal. per min.	Pressure delivery	Lb. per Sq. In.		h.p. output	Efficiency of pump
				Inj.	Net		
12:58– 1:14	1	3372	344.4	29.3	315.1	620	74.6
1:22– 1:37	2	3809	336.0	27.9	308.1	683	70.1
1:43– 1:58	3	3495	334.0	28.7	305.3	623	73.2
2:03– 2:18	4	3705	334.5	27.8	306.7	662	76.0
2:24– 2:38	6	3740.7	344.5	28.8	315.7	689	77.0

Immediately following the 24-hr. test for capacity.

65 Immediately after the close of the endurance test of 24 hours, a short test was run on each motor separately, which was continued long enough after uniform results were shown to obtain 12 to 15 readings. This test was run for the purpose of ascertaining whether there were deficiencies in any of the individual motors, and to meet the requirements specified in the printed specifications for the work, viz: that each pump should be free from defects, should have a capacity of 3,000 gal. per min. and an efficiency not less than 70 per cent. The results of these tests, Table 3, show that the individual pumps had an efficiency from 4 per cent to 6 per cent in excess of the average when operated together, and that the capacity for the specified discharge pressure was considerably in excess of the requirement of the specification. It is, I believe, generally the case that individual centrifugal pumps delivering water into a main singly show a greater efficiency by from 4 per cent to 6 per cent than the same pumps delivering together into a single main, due probably to less loss in eddy currents and friction head, etc.

GANSEVOORT STREET STATION TEST

66 The endurance test of the Gansevoort Street Station with all the pumps in operation was begun at 9:45 a.m., September 5, after the pumps had been operated for about 20 min. giving uniform results. The test was continued for 24 hr. The method of testing and the various observers were the same as for the tests at the South Street Station and the results are given in Tables 4 to 6.

67 For the Gansevoort Street Station the efficiency average for 24 hr. was 72.9 per cent, with a variation (excepting the first hour) of less than one-half of 1 per cent. It fell below 70 per cent during the first hour, which was due to the opening of an automatic relief valve on pump No. 2, which discharged some of the water into the suction before it had been metered. For that reason the efficiency during the first hour has not been considered in determining the performance of the pumps.

68 The least capacity during the eight consecutive hours when all the water pumped passed through the meters occurs from 10.45 a.m. to 6.45 p.m. The average capacity during this time is 17,419 gal. The average net pressure in pounds is 324.3 which is nearly 16 lb. in excess of the contract requirements. The average efficiency for the period above is 72.90 per cent.

TABLE 4 HOURLY AVERAGE OF READINGS OF DISCHARGE AND INJECTION GAGES ON PUMPS

GANSEVOORT STREET PUMPING STATION, SEPTEMBER 5 AND 6, 1908

HOUR	PUMP NO. 6		PUMP NO. 4		PUMP NO. 2		PUMP NO. 1		PUMP NO. 3		AVERAGE		NET PRESSURE LB.
	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	Disc.	Inj.	
9:45–10:30	342.4	24.7	342.6	25.6	344.8	24.9	346.9	25.2	345.1	25.3	338.4	25.1	319.3
10:45–11:30	347.4	24.7	347.1	25.7	345.4	25.2	348.4	25.4	346.4	25.2	346.9	25.2	321.7
11:45–12:15	348.9	24.9	348.9	26.2	347.9	25.7	349.4	25.9	347.9	25.9	348.6	25.7	322.9
12:45– 1:15	351.4	25.7	350.9	27.7	350.1	26.7	352.4	26.9	350.4	26.9	351.0	26.8	324.2
1:45– 2:15	352.4	25.7	352.4	27.4	350.9	26.7	353.4	26.7	351.4	26.9	352.1	26.7	325.4
2:45– 3:15	352.9	25.7	352.9	27.7	351.4	26.9	354.9	27.2	351.4	27.2	352.7	26.9	325.8
3:45– 4:15	353.9	26.4	353.9	28.2	353.4	27.7	355.4	27.4	353.4	27.9	354.0	27.5	326.5
4:45– 5:15	354.4	26.9	354.4	28.7	353.4	27.9	355.9	28.2	353.4	28.7	354.3	28.1	326.2
5:45– 6:15	349.9	27.4	349.9	29.7	350.4	28.7	351.9	28.9	350.9	29.2	350.6	28.8	321.8
6:45– 7:15	349.9	28.4	348.9	30.2	349.4	29.2	351.4	29.2	349.4	29.4	349.8	29.3	320.5
7:45– 8:15	351.9	28.9	350.9	30.7	350.4	29.7	352.4	29.2	349.9	29.9	351.1	29.7	321.4
8:45– 9:15	351.9	29.7	351.9	30.9	350.6	29.7	353.4	29.9	350.9	30.7	351.7	30.2	321.5
9:45–10:15	352.4	29.7	354.4	30.9	351.4	30.2	353.4	30.4	352.9	30.9	352.9	30.4	322.5
10:45–11:15	354.4	30.4	353.9	31.2	353.4	30.2	354.9	30.9	353.4	31.2	354.0	30.8	323.2
11:45–12:15	354.4	31.2	353.4	31.7	352.9	31.2	353.9	31.7	352.9	31.2	353.5	31.4	322.1
12:45– 1:15	353.9	31.2	352.4	31.7	352.9	31.2	354.9	31.7	353.4	31.2	353.5	31.4	322.1
1:45– 2:15	352.4	31.2	351.9	32.2	352.9	31.2	354.4	31.9	353.4	31.2	353.0	31.5	321.5
2:45– 3:15	350.4	31.7	349.4	32.2	349.4	31.4	351.9	32.2	350.4	31.7	350.3	31.8	318.5
3:45– 4:15	350.9	32.2	350.4	32.7	348.9	31.2	351.4	32.7	348.4	31.4	350.0	32.0	318.0
4:45– 5:15	350.9	31.9	350.4	32.4	350.4	31.2	352.9	32.2	349.9	31.2	350.9	31.8	319.1
5:45– 6:15	352.4	31.2	351.9	32.2	352.4	30.7	353.4	31.7	351.9	31.4	352.4	31.4	321.0
6:45– 7:15	350.9	29.9	349.9	31.2	349.4	29.9	351.9	30.7	349.4	30.9	350.3	30.5	319.8
7:45– 8:15	349.4	28.7	349.4	29.7	347.4	29.4	349.9	29.2	347.9	29.4	348.8	29.3	319.5
8:45– 9:45	348.1	27.4	347.7	28.9	346.4	28.0	348.4	28.0	346.4	28.5	347.4	28.2	319.2

Readings corrected for error of gage and to center of pumps.

69 The average capacity for the entire test is 17,867 gal. which was obtained with an average speed of 753.6 r.p.m.

70 Immediately after the completion of the endurance test of 24 hours duration, each pump was tested when operating alone for a period sufficiently long to obtain 12 to 15 readings after they had become practically uniform. These tests gave in every case an efficiency several per cent greater than that obtained when the pumps were all discharging into the same main.

CONCLUSIONS

71 It appears from the endurance test in each station that the capacity, efficiency and pressure exceeded the contract requirements by a large margin, and that during the endurance test no mechanical

TABLE 5 COMPUTATION OF PUMP EFFICIENCIES

GANSEVOORT STREET PUMPING STATION, SEPTEMBER 5 AND 6, 1908

Hour Beginning	r.p.m.	Total kw. per hr.	Total h.p. from motors (input)	Gal. per min.	Net pressure lb.	h.p. delivered	Efficiency per cent
9:45 a.m.	740	3671	4586.5	17107	319.3	3188.9	69.5
10:45	749	3589	4484.2	17310	321.7	3251.0	72.5
11:45	750	3591	4486.7	17290	322.9	3259.3	72.9
12:45	752	3591	4486.7	17280	324.2	3270.5	72.9
1:45 p.m.	752	3604	4502.9	17285	325.4	3283.6	72.9
2:45	753	3604	4502.9	17315	325.8	3293.3	73.3
3:45	753	3630	4535.3	17345	326.5	3306.1	72.9
4:45	753	3685	4604.0	17670	326.2	3365.0	73.1
5:45	756	3696	4617.9	17855	321.8	3354.4	72.6
6:45	756	3661	4574.0	17825	320.5	3335.2	73.4
7:45	754	3676	4592.9	17775	321.4	3335.2	73.3
8:45	753	3685	4604.0	17755	321.5	3332.5	72.8
9:45	755	3657	4569.0	17720	322.5	3336.2	72.9
10:45	755	3693	4614.2	17755	323.2	3350.1	72.7
11:45	756	3704	4627.9	17830	322.1	3352.8	72.6
12:45 a.m.	756	3753	4689.0	18195	322.1	3421.4	73.0
1:45	756	3760	4697.8	18310	321.5	3436.6	73.3
2:45	756	3735	4665.5	18315	318.5	3405.5	73.0
3:45	755	3725	4654.0	18290	318.0	3395.5	73.0
4:45	756	3743	4677.6	18315	319.1	3411.9	73.0
5:45	756	3784	4727.9	18330	321.0	3435.0	72.7
6:45	755	3747	4681.6	18315	319.8	3419.4	73.0
7:45	755	3723	4656.5	18255	319.5	3405.0	73.1
8:45	755	3722	4655.3	18189	319.2	3389.5	72.8
Average	753.6						72.9

Average efficiency, 1st period of 8 hr. = 72.9 per cent.
Average efficiency, 2d period of 8 hr. = 73.0 per cent.
Average efficiency, 3d period of 8 hr. = 72.9 per cent.

No. of cycles per sec. 9:45 a.m. to 2:45 p.m. = 25.00
No. of cycles per sec. 2.45 p.m. to 4.45 p.m. = 25.25
No. of cycles per sec. 4:45 p.m. to 6:45 p.m. = 25.50
No. of cycles per sec. 6:45 p.m. to 7:45 p.m. = 25.00
No. of cycles per sec. 7:45 p.m. to 9:45 p.m. = 25.25
No. of cycles per sec. 9:45 p.m. to 9:45 a.m. = 25.50

or electrical defects were observed. During the test of the South Street Station one of the pumps was stopped for half an hour to repair the motor insulation, while during the test of the Gansevoort Street Station no stop was made. The bearings in both stations were in perfect condition at the end of the test and the temperature of the motors not sufficiently high to interfere with the continuous operation for a longer period. Apparently the endurance test could have been continued indefinitely without injuriously overworking or overloading the pumps and motors.

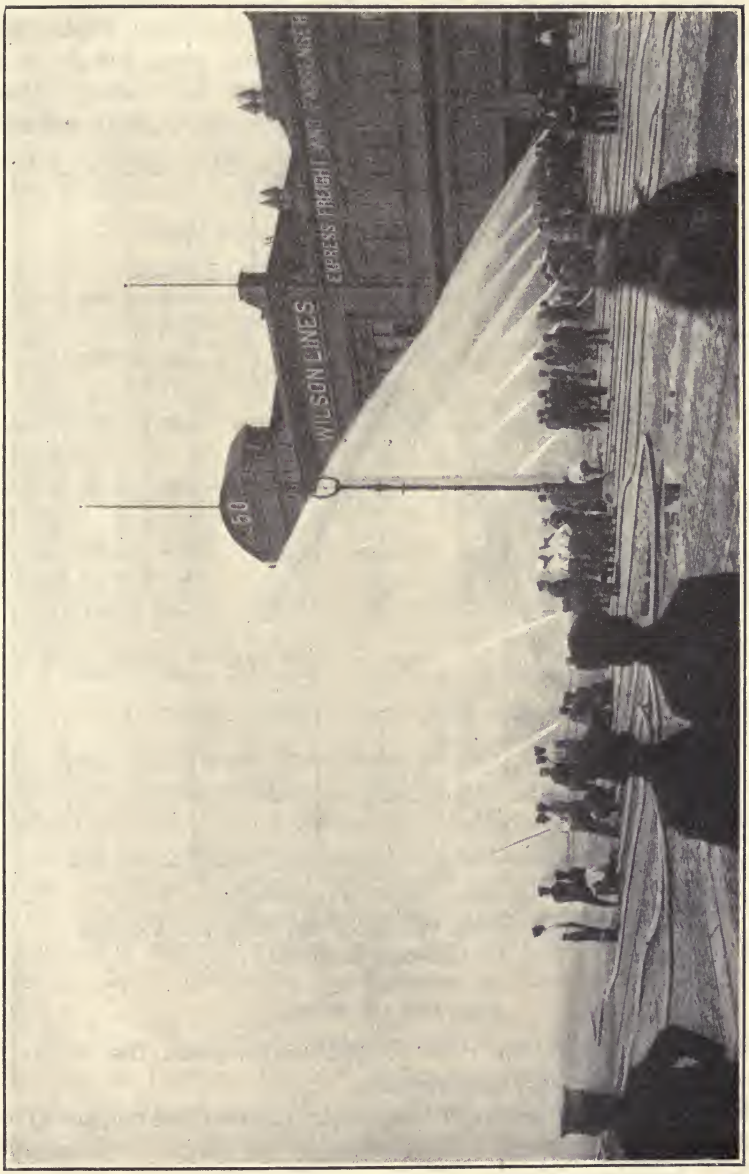


FIG. 7 VIEW SHOWING TEST OF FIRE SYSTEM

72 The specifications call for pumping sea water, which most authorities consider to be approximately 2.5 per cent heavier than fresh water. The effect of substituting sea water for fresh water would have been to reduce the capacity of the pump by about 2½ per cent for the same horse-power delivered by the motor, without sensibly affecting the efficiency. Because of the large capacity

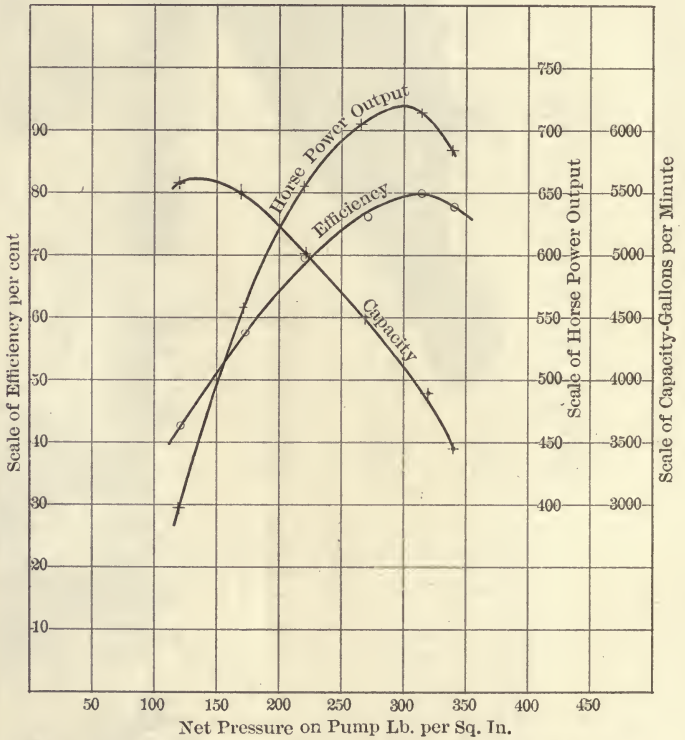


FIG. 8 CHARACTERISTIC CURVES OF THE PUMP FOR VARYING DISCHARGE-PRESSURES

RESULTS OF TESTS

shown by the pump, this does not materially affect the results in relation to the contract requirements.

73 The data and results of the tests at the two stations are given concisely in the tables. The efficiency is given as computed for each hour, and shows a slight variation which probably can be accounted for by changes in the amount of water leaking past the balancing piston. The individual pump tests at the South Street

Station show a variation in efficiency from 70 per cent to 77 per cent, and at the Gansevoort Street Station from 70 per cent to 79 per cent. This variation may have been due to the structure of the pumps but in my opinion is more probably due to variable leakage past the balancing piston or through the relief valves.

74 Pump No. 6 at the Gansevoort Street Station was tested with varying openings of the valve in the discharge pipe. The results are shown in the latter half of Table 6.

TABLE 6 TEST OF INDIVIDUAL PUMPS

GANSEVOORT STREET STATION, SEPTEMBER 6, 1908

Time	No. of pump	Elect. h.p. input	Gal. per min. Hg. Col.	Pressure delivery	Lb. Per Sq. In.		h.p. output	Efficiency of pump
					Inj.	Net		
10:05–10:31	1	916	3800	356.8	35.4	321.4	711	77.6
10:36–10:51	2	877	3800	350.8	35.1	315.7	700	70.8
10:54–11:12	3	920.5	3820	350.4	34.1	316.3	703	78.0
11:17–11:30	4	892	3751.4	352.5	35.6	316.9	695	77.7
11:37–11:53	6	899	3880	350.9	35.2	315.7	714	79.4
11:55–12:03	6	880.3	3457	376.1	36.0	340.1	686	77.9
12:03–12:07	6	929	4500	304.4	34.6	269.8	708	76.1
12:09–12:13	6	946	5070	255.6	33.6	222.0	654	69.4
12:24–12:28	6	952	5500	207.4	33.2	174.2	559	58.7
12:32–12:36	6	927	5588	155.2	33.2	122.0	397	42.8

Immediately following the 24-hr. test for capacity.

PRACTICAL RESULTS FROM THE NEW SYSTEM

75 The high-pressure fire system in New York, which was put officially into service on July 6, 1908, has been successfully operated at many fires, but it had a crucial test on January 7, 8 and 9, 1909, when it was brought into service for five simultaneous fires, three of them of more than the usual extent and activity, and one particularly so. Information upon the results attained with the system and the amount of water consumed was given by Chief Engineer I. M. de Varona and published in the Engineering News of February 11, 1909.

76 The fires occurred at Hudson and Franklin Streets, Hester Street and the Bowery, Houston Street and Broadway, Sixth Avenue and 17th Street, and Houston Street and the Bowery. The situation became so dangerous that every engine south of 37th Street, or 40 engines, were summoned, as well as a force consisting of 12 battalion chiefs and more than 600 men, but there was no need to use a single one of the engines.

77 As the violence of the fires increased, additional pumps were brought into service, so that at one time four pumps and motors were in commission at the South Street Station and three pumps at the

TABLE 7 SPECIFIED CHEMICAL ANALYSIS FOR PUMP MATERIALS

	Nickel steel	Medium steel	Steel forging	Steel casting
Parts of 1 per cent.....				
Phosphorus not to exceed.....	0.04	0.10	0.04	0.05
Sulphur not to exceed.....	0.04		0.04	0.05
Tensile strength at rupture, pounds	100,000	650,000	75,000	65,000
Tensile strength at elastic limit, pounds.....	65,000	32,500	38,000	32,000
Per cent elongation in 8 in.....	2	22		
Per cent elongation in 2 in.....	22		22	18
Contraction of area per cent.....	32		32	24
Carbon not less than	20 parts of 1%			
Nickel percentage.....	21 to 24			

Gansevoort Street Station, delivering 35,500 gal. per min. against an average pressure of 225 lb. at the pumps and 205 lb. at the hydrants. During the operation of the pumps 14,095,000 gal. were pumped as recorded by the meters, and the current used was 81,450 kw-hr., the cost of which was $1222.

DISCUSSION AT NEW YORK

PROF. GEORGE F. SEVER.[1] The electrical features of this installation are of much interest but the reasons for selecting that system which is now in operation should be given. In the discussion of this problem both alternating and direct-current power were considered for the operation of the motor-driven pumps, and alternating-current power was decided upon. The reasons for such selection I have noted herewith:

a Absolute simplicity, which is the key-note of the electrical end of this power installation.

[1] Professor of Electrical Engineering, Columbia University.

NOTE.—The high-pressure system was designed by I. M. de Varona, Chief Engineer of the Department of Water Supply, Gas and Electricity of New York. It was also constructed under his supervision. The construction of the electrical machinery was supervised by Prof. Geo. F. Sever as Consulting Engineer. The details of construction were in charge of Thomas J. Gannon, John P. Reynolds and Henry B. Machen, assistant engineers of the department. The machinery of each station was designed and erected by the Allis-Chalmers Co. of Milwaukee.

b Commutating apparatus and brushes are entirely absent.

c Induction motors provide very quick starting when it is necessary to operate the station on a fire signal.

d There is less expense for copper in the distribution system to insure continuity of service.

e The induction motor is a less expensive apparatus than the direct-current motor.

f With the induction motor there are absolutely no exposed live circuits in the station, as there might be with a direct-current apparatus. The final decision was for 3-phase service at 6600 volts and 25 cycles. It was decided that it would not be desirable to establish a power house to be operated by the city because it would be a municipal plant.

2 In order to insure continuity of service there is brought to each pumping station an independent feeder from each of the two Waterside stations of the New York Edison Company. There is also brought to each pumping station an independent feeder from the nearest substation of the New York Edison Company, as follows: to the Gansevoort Street station two feeders from the Horatio Street sub-station, and to the South Street station two feeders from the Duane Street station of the company. Hence there are really four independent sources of power supply for each pumping station, assuring practically no possibility of shutdown.

3 The contract for electric power for the Manhattan stations was let to the New York Edison Company. This contract provides for two payments, the first for a reservation of 3250 kw. capacity, of generating, distributing and controlling apparatus, available at either pumping station at an instant's notice, or practically without any notice at all. Thus four pumps can be thrown on with absolutely no notice to the New York Edison Company that they are to be used. For that reservation, and care and maintenance of the whole distributing system, the city pays about $63,000 per year, and the city also pays one and one-half cents per kw-hr. for all high-tension power used in each station.

4 Another stipulation in the contract may be of interest to engineers as it provides for the protection of the city. This stipulation is as follows: "If the contractor, under the terms of this contract, shall fail to maintain and deliver a continuous and uninterrupted supply of electric power when required, the contractors shall and will pay to the city the sum of five hundred dollars per minute for

each minute's interruption or delay of electric power supply after the power has been interrupted or delayed for three consecutive minutes." So, if they cannot deliver power after an interruption of three minutes, immediately a charge of $500 per min. is imposed and is deducted from the bills which the New York Edison Company renders.

5 The operation of both these stations is extremely simple. The handle of the oil switch is turned, throwing the 6600 volts directly on the stator of the motor. By turning a hand wheel, the motor is brought up to speed in less than 33 sec., and in starting the current is not supposed to exceed 150 per cent of the full-load current, which is 64 amperes. As far as I have observed the operation of the stations, there has been absolutely no trouble from the electrical end, no trouble with the feeder system, and none with the motors, and I think the City of New York has two plants which will give it for many years to come absolutely no trouble whatsoever.

Wm. M. White. The paper deals with questions in which I am directly interested. The methods employed in making the tests were probably the best that could have been selected. There is probably no more accurate method of determining the quantity of water delivered by a pump than by the venturi meter, especially when in the hands of an expert who is familiar with its workings. The venturi meter, as Professor Carpenter says, has been used for a number of years; it has been tested in various ways and proved to give accurate results. The power delivered to the pumps can be most carefully measured by electrical instruments.

2 The writer accepts without question the various efficiencies obtained and presented by the author, who states, calling attention to the variation in efficiencies obtained, that the individual observations do not agree as closely as he would like. I do not think Professor Carpenter should offer any apology as the results seem to agree very closely, and certainly are as accurate as are generally obtained on work of this kind. The efficiencies obtained on these pumps, though not the highest that have been obtained, are as high as is usual for similar conditions of head, capacity and speed. The designers of the pumps deserve credit for the performance shown by the pumps.

3 I am at a loss to find a reason for the variation in efficiencies of the pumps, as mentioned in Par. 65, where it is stated that individual pumps delivering water into a main singly show greater efficiency

than the same pumps delivering together into a single main. I assume, of course, that the variation in efficiency refers to the pumps when they are delivering exactly the same quantity against the same head at the same speed, whether working singly or in parallel. In the normal operation of pumps, it would be a fact that when one pump was operating from a suction main to a discharge main, the efficiency of that pump would be different from what it would be when working with another pump from the same suction main and discharging into the same discharge main, because the two pumps would usually be working against a higher head than when a pump was working singly. The increased head on the pumps would mean a decrease of capacity, and the increase of power demanded by two motors instead of one would mean a slight increase in line loss, which would again slightly decrease the speed and slightly change the conditions of operation for two pumps over that which would exist when one pump only was in operation. Of course, under these conditions, the two pumps would show different efficiencies, because the efficiency curve of a pump varies as its capacity and head.

4 I do not believe, however, that this is the condition to which Professor Carpenter refers. I assume that he has corrected for this difference, and has obtained from two pumps working in parallel the same capacities, heads and speeds as though one pump were in operation, and that under this latter condition he finds the difference in efficiency in the two pumps. If this be a fact, it is the most important point brought out from a designer's point of view.

5 I am at this time attempting to duplicate the conditions, to see whether the efficiencies are different under the same conditions of capacity, head and speed, as mentioned by Professor Carpenter.

George L. Fowler. A number of years ago I was associated with Joseph Edwards, who at that time had the contract for excavating the ship channel in New York Harbor, probably one of the first, if not the first, very large hydraulic engineering projects successfully accomplished by the contractor and to the satisfaction of the Government.

2 The ship channel leading from the Narrows down to Sandy Hook and out to sea, is about 15 miles long, and runs almost due south first, turning to nearly due east before reaching Sandy Hook, and passing through Gedney Channel to the sea. Cutting across it is

the Swash Channel, not used by any deep-draft boats. When the work was undertaken New York Harbor was shoal at two points on the Gedney Channel and the ship channel, where the water depth was a little less than 24 ft. The Government had a survey made and an estimate of costs based on material actually removed by the ordinary methods of dredging. Through the open space from Sandy Hook to Coney Island the whole lower bay is subject to all the winds coming in from the Atlantic on the east and across Raritan Bay, so that the water is nearly always rough. Two contractors had attempted the work by ordinary bucket dredging and both had failed.

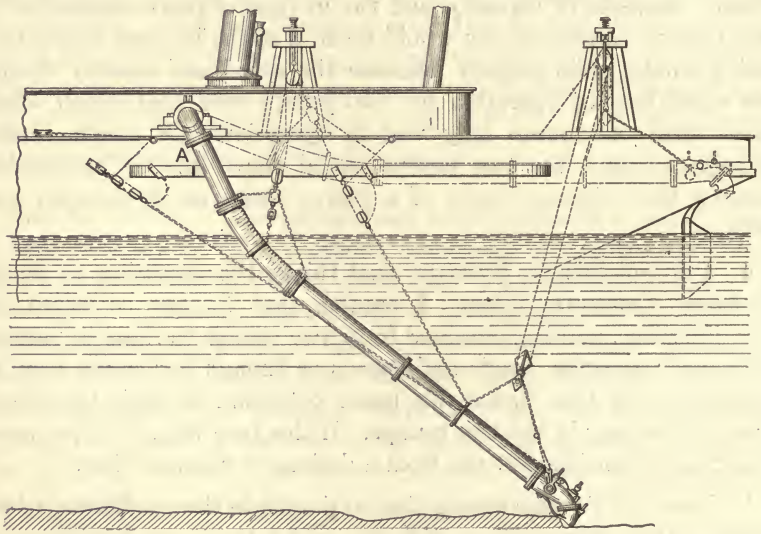


FIG. 1 HYDRAULIC DREDGER FOR DEEPENING SHIP CHANNELS

3 In the ship channel the material was sand and sedimentary clay, lying over hard sand; in the Gedney Channel it was gravel, shell and sand, for two feet overlying hard shingle. Hydraulic dredging was specially suited for this kind of work, and many kinds of material were removed from the channel besides the ordinary silt.

4 Three sea-going vessels were built for this work by the Joseph Edwards Company: the Reliance, the Advance, and the Mt. Waldo. Fig. 1 shows the general arrangement of the ships. At *A* is the long drag aft, where the pipe goes into the vessel and where the pumps are located, each driven by a 192-h.p. engine at 178 r.p.m. The suction and delivery pipes were 15 in. in diameter, with a shell of 40 in. The

pumps delivered 10,000 gal. per min. at a velocity of 1100 ft. The efficiency was thus between 65 and 70 per cent, although in later tests made by the Government, when nothing but water passed through the pipes, the efficiency rose to as high as 80 per cent.

5 The shoe used is a hook that drags along the bottom, chains being fastened to the vessel for this purpose. The vessel never stopped from morning to night, simply running out to sea, dumping, and coming back again to work.

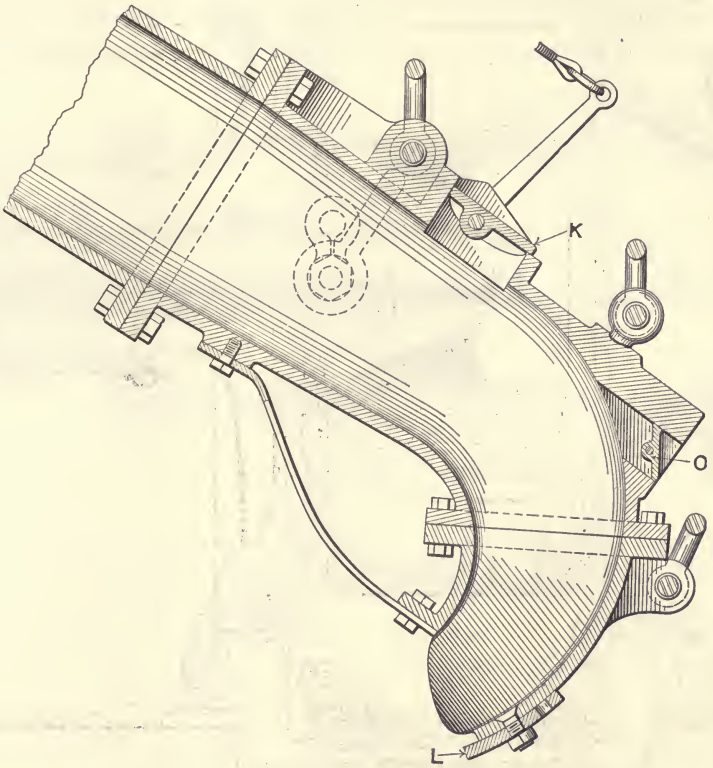


FIG. 2 DETAIL OF END OF SUCTION LINE

6 At the point *L*, Fig. 2, was the heavy shoe that served to dig into the mud and gravel. At *O* was a butterfly valve, kept open all the time to admit water above the drag to mix with the material raised. At the bottom *K* was another valve which could be opened in an emergency, in case not enough water was admitted at *O*.

7 The pump itself was of a plain centrifugal type, 40 in. in diameter, with vanes cut away at the center, as shown in Fig. 3. Because of this arrangement, the material would come in at *C* and out of the

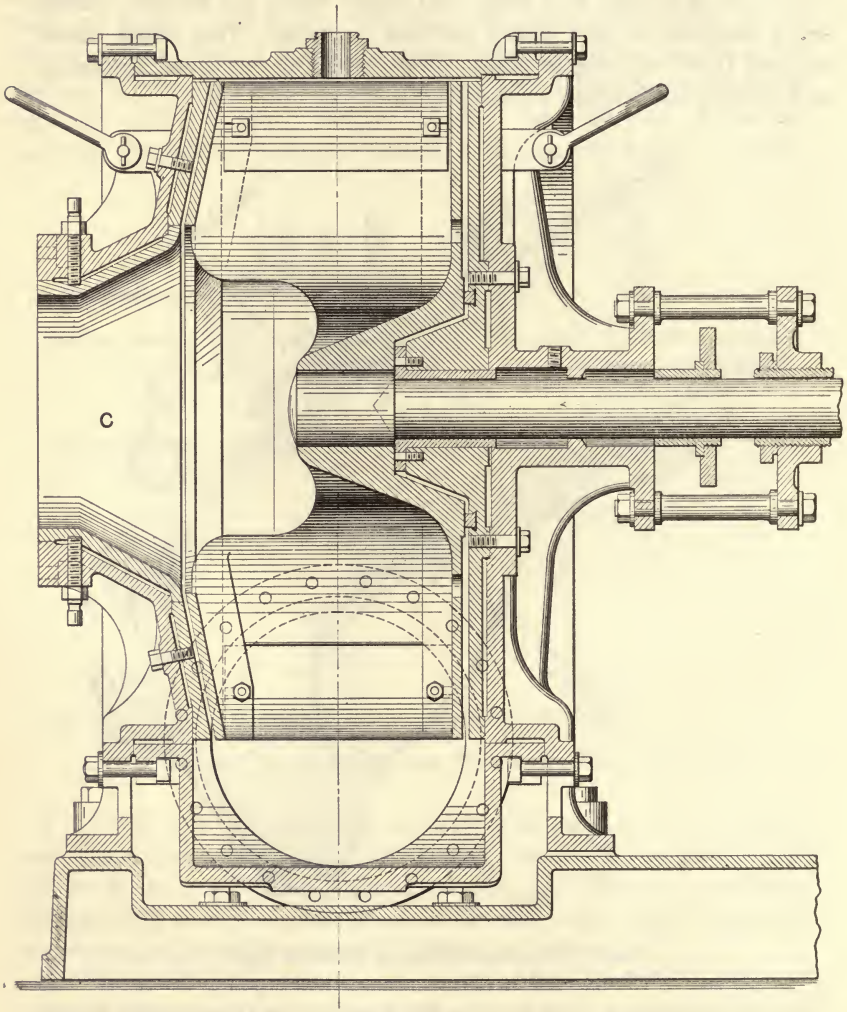


FIG. 3 SECTIONAL VIEW OF CENTRIFUGAL PUMP FOR DREDGING

vanes at the discharge, without damaging the pump when heavy substances were drawn in. The three vanes were made with wings

bolted on, and accessible from both sides. The thrust was taken up by the bearing at *T* (Fig. 4), the nuts marked *m* being screwed into a head carried by the bars *O*, bringing the thrust plates at the point *i*. The reason for threading the nut *m* was to adjust it to the vanes in proper relative position to the sides of the pump. That is a simple construction maintained ever since, with the exception that ball bearings are now used.

8 Although the pumps were originally intended to take water and other loose material, such as sand and gravel, they proved capable of lifting practically anything that came in their way. The three following specimens are interesting as showing the pumps' lifting power:

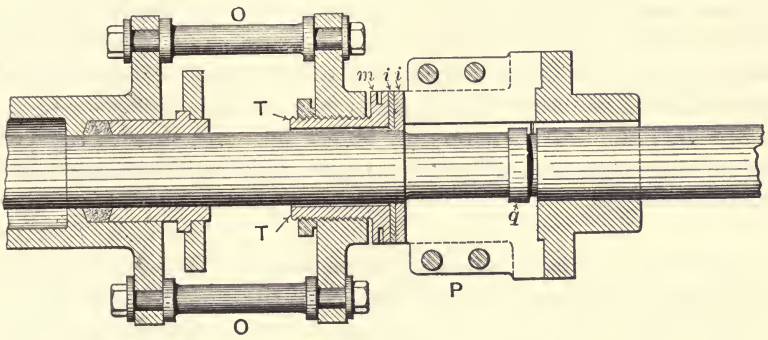


FIG. 4 DETAIL OF THRUST BEARING OF PUMP

a A piece of shaft weighing 70 lb. raised and passed by a 15-in. dredging pump; improvement of New York Harbor, Steamer Reliance.

b A piece of tree root raised and passed by a 12-in. pump from 14 ft. of water at Miami, Fla.; Florida East Coast Railway Company improvements.

c A piece of pig iron measuring 11½ in. by 4¾ in. by 3¼ in. and weighing 35 lb. raised and passed by an 8-in. special cataract wrecking-pump from 15 ft. of water from the wreck of a canal boat sunk at Puas Dock, Yonkers, N. Y.; by Baxter Wrecking Company, New York.

9 For hydraulic dredging, the Government pays by the scow load and gets what is excavated. In ordinary hydraulic dredging, like that in the ship channel, about 15 per cent of the pump discharge was solid matter. About 40 per cent in excess of the amount deposited

in the bins went overboard with the overflow, and was carried out to the flats at the sides by the cross currents, which also carried the loose material stirred up by the drag. The result was that the Government obtained an excavation about 70 per cent in excess of what would have been obtained had all of the material removed from the bottom been caught in the bins. This, of course, greatly reduced the actual cost of the excavation. For example: the last contract made on the ship channel was at the rate of 16⅞ cents per yard, while with the allowance indicated, above the actual cost per yard—channel measurement—it was about 11 cents.

10 As for the time of loading, some records indicate that this ship, 157 ft. long and with a capacity of 650 cu. yd., was loaded in 48 min.; there are also records of its being loaded at the rate of 16 cu. yd. per min., of solid matter placed in the bins; and records of its taking out to sea nearly 4000 cu. yd. per day. The vessel was worked in all kinds of weather, even when tackles had to be used to board her; and yet the ship was taking her load steadily. Except in the case of an actual breakdown the work could be carried on for 16 hr. per day.

John H. Norris. In a pumping plant of the character described, this type of equipment seems in the present state of the art the most suitable that could have been selected. I would like, in this connection, to call attention to another type of installation for service of this kind, though not on so large a scale, which appeals to me as being more desirable than the electric-driven centrifugal pumping plant taking its power from a public utilities company.

2 At Coney Island was installed the first plant operated by the City of New York for fire protection by means of water delivered into mains under high pressure, with the idea of taking care of a restricted area where there was great danger from fire.

3 This plant consists of three 150-h.p. three-cylinder, vertical gas engines direct-connected to triplex pumps, each unit capable of pumping 1500 gal. per min. against a pressure of 150 lb. These engines take their fuel from the mains of the local gas company and can be arranged if necessary to run on gasolene. They are installed in a building on city property and are arranged to take their water supply from the city mains or from Coney Island Creek, within 50 ft. of the pumping station. The engines are started with compressed air, and the three units can be started up in less than three minutes. On every occasion they have been found ready for service whenever

the demand was made upon them. The cost of this pumping station was as follows:

Building	$10,000	
Equipment	37,000	
		$47,000

The annual operating expenses are:

Labor	$13,140.00	
Supplies and Repairs	897.27	
Fuel	150.00	
		$14,187.27

4 By comparing the foregoing figures it will be evident that for service smaller than is required in the City of New York, the gas-engine-operated triplex pump gives an economical equipment that can be allowed to stand idle for any length of time and yet be ready for instant service.

5 New York City pays the New York Edison Company an annual charge of $90,000 for the privilege of calling for sufficient current to operate the equipment at any time. This item capitalized at 5 per cent would pay for a good-sized gas-engine plant.

6 The following data were taken from the capacity tests of the Coney Island units:

Duration of test	14 hr.
Average piston speed of pump	90.3 ft. per. min.
Total head pumped against	156.5 lb.
Average pump horsepower for each unit	142.2 h.p.
Average gas consumed per hour for the 3 units	8914.0 ft.
Average capacity	4512.0 gal. per min.
Slip of pump	3.45 per cent
Average efficiency of pumps	82.00 per cent

J. R. Bibbins. Although Professor Carpenter's paper deals primarily with multistage pumps, I wish to direct attention to the question of motive power, upon which the success or failure of the system practically depends. We have seen excellent examples of two systems diametrically opposed in regard to power supply—the electrical and the gas-driven system. Under certain conditions, both are extremely serviceable. The first high-pressure installation on a large scale, in this country, was the gas-driven system at Philadelphia. Although I have not had an opportunity to follow the results of that station for the past two or three years, the results obtained and published for

the first year or so showed that such a system of gas-driven pumps merits every consideration.

2 *First as to the security of power supply:* In Philadelphia the Delaware Avenue station receives its gas supply directly from a 24-in. trunk main running between two very large gas holders, located in different parts of the city. Roughly, the pipe line measures four miles in length, its capacity constituting a considerable reserve in itself, if both the holders were unavailable. There is no intermediary apparatus whatever between the pipe line and the engine; that is, the plant may draw directly on these two large holders of several million cubic feet capacity. This constitutes a very safe and reliable source of motive power which can hardly be paralleled except, perhaps, by the situation in the New York electric service, where there are so many stations to draw from.

3 In this connection, I would like to ask whether it is at present possible to utilize the storage battery capacity in the various sub-stations for reserve service at the high-pressure pumping station. It is stated that the storage batteries are available for reserve in emergencies, such as discontinuance of the main high-tension current supply. I am under the impression that an inverted rotary requires a direct-driven exciter to maintain a definite frequency and prevent racing. Without special controlling apparatus, this inversion would be impossible in the ordinary sub-station equipment. Possibly special provision has been made in the New York systems, in which case, the security of power supply is certainly beyond criticism. In other words, would it be possible to invert the synchronous converters on short notice?

4 *Second, quick starting:* It seems to be a fact that a large part of the minimum time required for the starting of a fire-service station is consumed in the operation of the motor-driven by-pass valves. In Philadelphia these valves are operated from an independent supply, as in New York, and at least fifteen seconds are required to close them; whereas the engines are brought up to speed within half a minute from the time the signal is given, the remaining time being usually consumed in closing this motor-driven valve.

5 The various tests of the Philadelphia plant showed that each of the units could be readily put on the line in well under one minute. It is an interesting fact that the original underwriters' tests specified the time limit as twelve minutes for the starting of the first three units, whereas the whole station can be started in that time, and has been started in seven minutes.

6 During the 36 days of preliminary service trials of the Philadelphia station, out of one hundred alarms given, only four misses were made in getting any of the eleven units started. In not a single instance has the station, as a whole, failed to respond to the service, at least during the period over which my observation extended. This has been accomplished with the regular operating force of three men.

7 *Third, in regard to the cost of service at Philadelphia;* The only data on a large fire available, are those of the fire in the Coates Publishing House, which lasted about nineteen hours. The average cost for pumping was about six cents per thousand gallons, including gas, wages and supplies. The cost of the large East Side service, cited in the paper, is about nine cents for power alone, and I think this does not include the readiness-to-serve factor. On the other hand, it is patent that the cost of service in either the gas or the electrical station is relatively unimportant. The main desideratum is reliability.

8 Finally, I desire to advance an argument for the development of a new type of pump unit, namely, a high-speed gas-driven centrifugal pump. Some time ago, in connection with water-works service, I found great difficulty, even with the present high-speed single-acting gas engine, in matching engine speeds with those required in centrifugal pump work However, for the pressure necessary in water-works practice, about 125 lb., one or two sizes of engines were found to be directly applicable to multistage pumps, with fair proportion of parts and good efficiencies. It seems possible to adopt a modified type of gas engine which would permit the direct connection mentioned.

9 This modification would naturally follow along lines of short stroke and high piston speeds with perhaps four cylinders. The engines at Philadelphia were designed with a piston speed of but 730 ft. per min. with a 22-in. stroke. This might be increased to 1000 ft. per min. without exceeding present-day limits, especially for units designed for occasional service. Such a unit would find immediate application in many industries and would combine the high economy of the gas engine with the simplicity of the centrifugal pump. The efficiencies shown by Professor Carpenter place the centrifugal pump in a position of closest competition with reciprocating pumping units.

J. J. Brown. I recently made a series of tests on three 6-in., 8-stage centrifugal pumps, each designed for 1000 gal. per min. and 560 lb. pressure at 1200 r.p.m. One of these pumps gave an efficiency from wire to water of 71 per cent, or a pump efficiency of 76 per cent.

regret that Professor Carpenter did not give the results of his tests on the New York fire-service pumps at lower capacities. All of the tests were made at capacities considerably in excess of that for which the pumps were designed and they apparently show their best efficiency at approximately 25 per cent over the normal rating. This increased efficiency at excess capacity seems to be apparent in several recent tests made on high-lift centrifugal pumps. The 8-stage machines previously referred to give their best efficiency at 1300 gal., or about 30 per cent over rating.

2 Mr. White has raised a question as to the difference in efficiency between the New York fire-service pumps working in multiple and as separate units. I think this is occasioned by the variation in capacity of the pumps when working together on a common suction and discharge line. I have found it rather difficult to balance two centrifugal pumps on a common discharge, and pitot tube tests indicate in almost every case a considerable difference between the amounts of water handled by the individual units under these conditions.

3 I have in mind one installation on fire service, where the pumps were called upon to deliver against the maximum pressure for which they were designed and it was only with considerable difficulty that we were able to cut in additional units. I think that if venturi meters or pitot tubes had been placed on the discharge of each of the five pumps when they were working in multiple, a difference in capacity of the several units would have been shown, which would account for the difference in efficiency observed when the pumps were working individually and not in multiple.

George A. Orrok. At the time of the award of contract for these fire pumps, the New York Edison Company was obtaining proposals for centrifugal feed pumps—a somewhat similar service—and eight 1000-gal. 300-lb. pressure five-stage pumps were purchased. There was no attempt to obtain a high guarantee for efficiency, but the builders did state that under the above conditions an efficiency of 65 to 68 per cent would be obtained. These pumps were of the Jager type and under test showed an efficiency of about 68 per cent.

2 Fig. 5 shows that the high-pressure fire-service pumps are of the Kugel-Gelpke type and should be a trifle more efficient because of smaller friction and leakage. Seventy-one per cent seemed a very high efficiency and many doubts were expressed regarding the fulfillment of the guarantees. The extreme figure of 79 per cent obtained is probably the result of careful design and extra good shop

work and I believe has not been excelled. That this figure came as a surprise may be explained by the fact that most centrifugal pumps are stock pumps and not specially designed for the work they have to do. Pump manufacturers have been more concerned in getting a line of patterns that will suit standard conditions than in developing a line of pumps and system of patterns capable of doing the best work.

3 As a centrifugal pump is a mixed-flow or Francis reaction turbine reversed, similar care in design and construction would probably give efficiencies similar to those of the best makes of reaction turbines, which approximate 90 per cent.

FREDERICK RAY. The difference in efficiency of the units operated individually from that obtained when several were operated in parallel might be due to the different rates of flow through the venturi meters under the two conditions. With one pump operating, this flow would be low and the mercury column reading would be but slightly over an inch, so that with a given error of observation the percentage of error would be much greater than with two or three pumps discharging through the same meter.

2 Professor Carpenter here replying that the pipe connecting the two meters was open all the time, Mr. Ray continued:

3 This would equalize the flow in the meters, so that the mercury column reading when the whole station was running would be about 6¼ times the reading with one pump. It has not been my experience that parallel operation of a number of pumps has any tendency to decrease or otherwise change the efficiency obtained when operated individually. The efficiency should be the same, and in this case, as the pressures were taken at each pump, any losses in the piping system due to parallel operation would be external to the gages and would not show in the calculations. If the pressure had been taken at the discharge of the whole system, losses in the piping would affect the results.

4 Many pumps are running under similar conditions, at the efficiencies given. I have myself obtained efficiencies of 80 per cent and higher, but I do not rely as much on them as on some a little lower. I am now testing a 6-in., 2-stage underwriter pump, having a normal capacity of 500 gal. per min. against 100 lb. pressure, which has developed a maximum efficiency of 73 per cent.

5 I think the centrifugal pump is the ideal one for fire service, not only on account of its simplicity and reliability, but also on account of its characteristic increase in capacity as the pressure is

reduced. Thus, the 500-gal. underwriter pump referred to will discharge 870 gal. per min. at 60 lb., or enough for four streams at this pressure. It will give three streams at 90 lb., two streams at 110 lb. and one at 117 lb.—all at constant speed without any regulation whatever.

6 The City of Toronto has recently issued specifications for centrifugal pumps for their general municipal water supply, among which are several fire pumps capable of discharging against 300 lb. pressure. These pumps, however, are to be equipped with variable-speed induction motors, the pressure regulation being obtained by speed variation. This is superior to throttling regulation from the standpoint of current economy and in the case of the New York installation a considerable saving could be made by this means, as most of the fires can be handled with 200 lb. pressure or less.

H. Y. Haden. A somewhat unusual result obtained from this type of pump is that as the total head continues to increase beyond a certain point, the capacity falls off, with the result that the capacity curve, as given in Fig. 8, shows a backward tendency. It will be interesting to get the explanation of this.

2 There is unquestionably a large field in fire protection for steam-turbine-driven centrifugal pumps, and it is to be hoped that the Fire Underwriters will officially accept this type of fire protection unit. I believe that a properly designed centrifugal pump, for high speeds and of few stages, can be used to great advantage when direct-connected to high-speed turbines.

Thomas J. Gannon.[1] It was decided to use electricity as power for the pumping stations, because the first cost of installation, yearly cost of operation and maintenance and fixed charges were estimated to be lower, taking into account the intermittent service. The construction and operation of a steam plant were entirely out of consideration and the choice lay between gas-engine-driven and electric-driven pumps receiving power from outside sources.

2 It was estimated that gas operation of plants equal in capacity to the present electrically driven plants, would involve a fixed charge of $50,000 a year, in addition to the cost of the gas actually consumed. The question as to who should build and maintain

[1] Engineer, Dept. Water Supply, Electricity and Gas, Manhattan Borough New York.

the necessary large gas mains, the cost of which would approximate a million dollars, was not definitely settled. That the cost of a gas-engine-driven pumping plant would have been approximately double, both for machinery, building and area of land to be purchased, is borne out by the actual costs of the installations in Manhattan and at Coney Island.

3 The capacity of the gas-operated Coney Island plant is 4500 gal. of water per min. against a head of 150 lb. per sq. in. The combined capacity of the two pumping plants in the Borough of Manhattan, as originally laid out, was 30,000 gal. per min. against a head of 300 lb., with provision in each station for three additional pumping units of a capacity of 3000 gal. each, making a total combined capacity of 48,000 gal. per min. against 300 lb. pressure. On actual test, however, the capacity of the pumps was approximately 20 per cent greater than the designed capacity.

4 Furthermore, the flexibility of this type of pump permits of an increased discharge at lower pressures, which gives a capacity of approximately 5500 to 5600 gal. per min. for pressures between 150 and 200 lb., or a combined total capacity of 55,000 gal. per min. against 200 lb. pressure. This corresponds to the pressure at which the station is operated for most fires. In other words, the water horsepower of the electric-driven as compared with the gas-engine-driven plant is approximately in the ratio of 20 to 1.

5 The cost of the machinery in the Coney Island plant was approximately $37,000, and the cost of the building approximately $10,000. The cost of each of the two Manhattan pumping stations complete, exclusive of land, was practically $240,000. The first cost of installation of the gas-engine-driven plant is therefore more than double the first cost of installation of an equivalent electrically-driven plant, in the city of New York.

6 The high-pressure fire-service pumping stations went into official operation on July 6, 1908. It was at first decided to put the stations in service only when called on by the fire department, and up to and including November 20, 1908, the pumping stations were called upon to go into actual service for but 17 fires. On that date, the method of operation was amended so that the pumping stations are put in service in response to every alarm in the high-pressure district, and continue in operation awaiting instructions from the fire department. Under this system, from November 20 to December 31, 1908, the pumps responded to 116 first alarms. From the best available information, water was used in 55 instances, making a

total of 72 fires for which the high-pressure service had been used up to that date.

7 To insure readiness for service at all times, daily tests are made, of at least half an hour's duration, unless the station has been in actual operation during the preceding 24 hours.

8 During the first quarter of 1909 the number of alarms received was 239, and water was taken from the station for 125 actual fires. The total amount of water pumped was 17,840,000 gal., and 145,900 kw-hr. was consumed. It was on January 7, 8 and 9 of this quarter that the three large simultaneous fires mentioned in Par. 75, occurred, for which over 14,000,000 gal. of water was pumped, leaving about 3,800,000 gal. for the balance of actual fires occurring during the quarter. For these three simultaneous fires more than 81,000 kw-hr. was consumed while the total consumption of power for the quarter for all fires and testing purposes was but 145,900 kw-hr.

9 As to why a pump running singly develops a higher efficiency than when running in conjunction with several others, it is observed that pumps of the same type do not necessarily develop their best efficiency at the same speed and pressure. The pump running singly will naturally develop a pressure which corresponds to its own design, but when working in multiple, it will have to adjust itself to the common pressure.

10 As to reliability I have neither seen nor heard of any time when any one of the ten pumps installed in the Borough of Manhattan has failed to respond instantly when called on for service and to develop the full pressure on the system within one minute's time. At no time in service have the pumps shut down of their own accord.

HENRY B. MACHEN.[1] Among the many difficulties encountered during the construction of the distribution system, perhaps the greatest was that due to the congested sub-surface of the street, which was a source of continual extra expense to the contractor, and of worry to the man in charge of selecting the location for the excavation of the trench.

2 The intersection of Sixth Avenue and Fourteenth Street may be cited as an example, since complete notes are available, due to the station excavation for the Hudson Tunnels. Here there were nine gas mains east and west, and nine north and south, belonging to

[1] Engineer, Dept. Water Supply, Electricity and Gas, Manhattan Borough, New York.

four different companies; two water mains in each direction; sewers and their connections on each side of the street; five Edison duct lines, and five duct lines with large manholes belonging to the Consolidated Telegraph and Electric Subway Company or the Empire City Subway Company; the conduits and banks of ducts of the Fourteenth Street and the Sixth Avenue trolleys; and lastly, the columns of the elevated railroad with their deep foundations.

3 Through this network the high-pressure main had to be so laid that the construction of the Sixth Avenue tunnel would not require it to be relaid. The excavation was carried on by tunneling, with here and there an opening through which the earth could be hoisted, using a pail let down by a rope. The pipe was lowered into the trench some distance up the street and pulled through, piece by piece, inspection of the running of the joint and caulking being almost impossible, since the space admitted but one man at a time after the pipe had been hauled in.

4 This condition existed at nearly all intersections of the main thoroughfares, such as Broadway, Sixth Avenue, Fifth Avenue, the Bowery, etc., and accounts for the high cost of laying the mains, averaging about $11 per ft. complete.

5 The second great difficulty encountered was in obtaining the prescribed test, which called for 450 lb. pressure per sq. in. to be held for 10 min., during which time the leakage was measured.

6 The system contained about 40,000 castings, 30,000 being straight pipe, tested at the foundry to 650 lb. The specials were not tested. All these castings, as already stated, were tested in the ground to 450 lb., the mains being under pressure in sections about one block long, between gates.

7 During the eighteen months the system has been in service, there have been but three breaks in the mains, all three in castings which had been subjected to the foundry test of 650 lb., two breaking at 150 lb. and the third at 300 lb. pressure.

8 To overcome the danger should a break occur during a fire, the proposed extensions to the distribution system now under contract, amounting to about $1,500,000, are laid out on what the department calls the duplex system. This method of overcoming the difficulty was first suggested by Mr. Blatt, assistant engineer of the high-pressure bureau. It consists of laying two entirely independent systems of mains and hydrants in alternate streets, the hydrants of one system being painted red and the other green. The mains are so laid out that at nearly all intersections of streets hydrants of both colors are available.

9 Should a break occur in either system, the operator at the pumping station would at once know in which system the trouble was located by looking at the venturi meters, and by throwing a switch he would start the closing of two electrically driven valves, separating one system from the other. Hydrants would then be available and in service pending the location and isolation of the damaged section.

10 The section now in operation was designed to give 20,000 gal. per min. on any one block with a loss due to friction from pumps to hydrant not to exceed 40 lb. The duplex extension will give the same results, and should either half be out of service by an accident, there will still be available at the same location 10,000 gal. per min., with a loss from the pumps to the hydrant in the most unfavorable location not exceeding 50 lb.

Richard H. Rice. This paper shows that the installation described was made after the most careful study and a very intelligent choice of the types of apparatus to be used. The choice of the centrifugal pump for the work described is thoroughly justified by its simplicity and by the efficiencies obtained. The centrifugal pump is today the popular means of producing pressure for emergency fire purposes, as in the fire boats of New York, Chicago, Duluth and San Francisco, and the new high-pressure service of San Francisco. In San Francisco twelve of these pumps are now being installed, four on fire boats and eight for an auxiliary fire installation. On the fire boats centrifugal pumps are particularly adaptable as they can be run in series or in parallel. In parallel they give 150 lb. pressure, and in series the pressure is doubled. This pressure is particularly valuable where walls have to be battered down, or streams thrown long distances.

2 The choice of alternating current as the source of power, in view of the unlimited supply of current existing and the duplicate means of conducting it into the station, is also justified. In cases where electricity is not so available as it is in New York, steam turbines are being installed, and they offer advantages over the gas engine, where maximum reliability is considered.

3 As an emergency installation pure and simple, I think the installation mentioned in the paper can be still further simplified. I believe the speeds chosen for operating the pumps are too low, and that the pumps contain too many stages. I have had occasion to make extensive researches in centrifugal pump design with special

reference to operation at steam-turbine speeds, and have found that they can be operated at high speeds with a smaller number of stages, giving efficiencies comparable with those obtained here, although the question of efficiency is subsidiary to reliability for this service. Pumps for this service should be designed with two or three stages at the most, and with considerably higher speed.

4 Pumps can also be designed without balancing pistons, which are undesirable from the viewpoint of possible interruption of service. An inspection of Fig. 5, illustrating the construction of the pumps, will show that the balancing pistons used are quite liable to damage if water containing sand or other impurities is used, and this damage would very probably result in stoppage of the pump when it is badly needed. The use of balancing pistons is unnecessary in such emergency apparatus and should be avoided.

C. A. Hague. A question has been asked several times with reference to the results of tests of efficiency on centrifugal pumps operating singly and in multiple or group. Professor Carpenter has given the very plausible explanation that the difference in efficiency in favor of the pumps running singly is probably due to the presence of eddies and disturbances in the pipes when the pumps are operating together and the absence of such eddies and disturbances when only one pump is at work. In my experience in installing pumps and condensers singly and in groups I have found them extremely sensitive to each other in operation, both in taking in and discharging the water, when more than one pump is working on a line.

2 In the Manhattan stations, it seems to me that the suction or inlet pipes and the discharge pipes are coupled too closely for best efficiency; and also that the inlet pipe close to the pumps is not large enough for operation in multiple, although perhaps ample for a single pump when the water is undisturbed by the draft and discharge of several pumps. I have experimented considerably in that line, and have found that a comparatively large body of water next to the pumps on the suction side will materially ease the machines in their performance. The idea is to come up to the building with a normal supply pipe, and then enlarge it very considerably just where it enters the building, providing the inlet pipe with a good-sized air chamber wherever possible. I have tried this several times with excellent results.

3 Mr. Brown mentioned the difficulty of cutting in with a second

pump where the first pump was already running, a difficulty which I think is also due to too close connections along the inlet and outlet lines and a cramped condition generally. Of course, a disturbance in the water column and in the hydraulic horsepower would unbalance the electric power to a certain extent, perhaps not much, but the total disturbance may very easily result in the loss of several points in the efficiency.

4 Considering the fact that the city pays by the kilowatt-hour for its electric current as per switchboard reading, it would be no more than proper to state the efficiency of the machine as a whole, and not exclusively upon the basis of motor efficiency obtained in the shop of the makers a thousand miles or so away. In this case when 100 h.p. in current is supplied to the switchboard, the motor has shown an output by a competent test of 93.2 h.p. (Par. 37), the balance of 6.8 h.p., charged against the city in the power bills, being lost in heat and friction. Then, all that is charged against the pump is 93.2 h.p. The 67.57 h.p. shown by the pump for each 100 h.p. at the switchboard indicates only 67.57 per cent total efficiency, although the 67.57 h.p. indicates 72.5 per cent efficiency of the power delivered by the motor. I have tested several centrifugal pumping plants of various sizes and powers, and the total efficiency generally shows from 64.5 per cent to about 68 per cent and very seldom above the latter figure.

5 Mr. Bibbins touched upon the possibilities of utilizing the centrifugal pump for waterworks service, but upon investigation he would find a vast difference between emergency service, where operating economy counts for little in the face of great danger from fire, and the steady and necessarily economical service required for the continual pumping in waterworks stations. To show how deceptive a portion of the truth may be, a case is cited where a pumpage of a capacity of 10,000,000 gal. per day against 110 lb. load could easily be accomplished with displacement steam machinery by an expenditure of $10,000 per annum for coal. But an attempt to drive centrifugal pumps by electricity resulted in a cost for electrical power, at $6.50 per 1,000,000 gal., of $23,725 per annum; showing a difference in favor of displacement steam machinery equal to 5 per cent per annum on $275,940. There is no conceivable difference in cost of machinery, buildings, maintenance, attendance, or anything else, that would justify such a preference for electricity and centrifugal pumps over steam and displacement pumps. Note the following figures:

10,000,000 gal. daily, against 110 lb.	440 pump-h.p.
120,000,000 steam duty with 8 lb. evaporation in the boilers, coal at $2.50 per net ton delivered	$9928 per annum
Electric power at $6.50 per 1,000,000 gal. against 110 lb. means 3,650,000,000 gal. per annum at $6.50	$23,725 per annum
The difference in cost for the element of power is $13,797 per annum, which at 5 per cent would capitalize at	$275,940

6 The steam-driven, reciprocating, displacement pumping engine can show a mechanical efficiency from the power put in through the throttle to the water-horsepower of the pumps, as high as 96 per cent, never as low as 90 per cent, under the above conditions. The centrifugal pump when steam-driven has a corresponding efficiency of about 65 per cent, and when electrically driven of about 67 per cent. A comparison of tests is given in Tables 1 and 2 in which it will be seen that the steam plant saves enough to pay 8.6 per cent on its entire cost.

TABLE 1 COST OF OWNING AND PUMPING WITH HIGHEST TYPE AND CLASS OF STEAM PUMPING MACHINERY

ONE UNIT, STEAM-DRIVEN, RECIPROCATING, DISPLACEMENT MACHINERY, CAPACITY OF 25,000,000 GAL. AGAINST 87 LB.

Pump horsepower	870
Boiler horsepower for triple-expansion vertical pumping engine	450
Engine house and foundations and engine foundations Boiler house and foundation, boiler foundations, chimney, etc Vertical triple-expansion pumping engine 450 h.p. of boilers Building for coal supply	$150,000

CHARGES AGAINST PLANT—PUMPING ENGINE

Interest	4 per cent
Sinking fund	5 per cent
Depreciation	2 per cent
Oil waste, etc	1 per cent
Total	12 per cent

CHARGES AGAINST PLANT—BOILERS

Interest	4 per cent
Sinking fund	5 per cent
Depreciation	5 per cent
Total	14 per cent

3 engineers. 6 firemen. 3 oilers.
Coal at $2.10 per net ton

SUMMARY FOR STEAM RECIPROCATING MACHINERY

Coal per annum	$11,957.40
Wages per annum	9,900.00
Capital charges on engine	13,920.00
Capital charges on boilers	1,260.00
Capital charges on buildings	1,548.00
Total charges per annum	$38,585.40
Cost per 1,000,000 gal	$4.11
Cost per horsepower	43.16

TABLE 2 COST OF OWNING AND PUMPING WITH HIGHEST TYPE ELECTRO-TURBINE PUMPING MACHINERY

ONE UNIT, ELECTRIC-DRIVEN, CENTRIFUGAL MACHINERY, CAPACITY 25,000,000 GAL. AGAINST 87 LB.

Pump horsepower	870
Two-stage, electric-driven centrifugal pump Engine house and foundations and pump foundations Transformer house and foundations Transformers, lightning arresters, conductors, controllers and auxiliaries	$43,750

CHARGES AGAINST PLANT—PUMPING MACHINERY, ETC.

Interest	4 per cent
Sinking fund	5 per cent
Oil, waste, etc	1 per cent
Depreciation	2 per cent
Total	12 per cent

3 Engineers. 3 Extra men
Electric current, $4.50 per 1,000,000 gal.

SUMMARY FOR ELECTRO-TURBINE MACHINERY

Electric current per annum	$41,062.50
Wages per annum	5,700.00
Capital charges on machinery	4,314.00
Capital charges on buildings	468.00
Total charges per annum	$51,544.50
Cost per 1,000,000 gal	$5.64
Cost per horse power	59.24

THOS. J. GANNON. In reply to Mr. Hague I will read the conditions which occurred on the evening of January 7, when both pumping stations were put to a crucial test:

7.22 First alarm, Hudson and Franklin Sts.
7.28 Second alarm, Hudson and Franklin Sts.
7.29 Third alarm, Hudson and Franklin Sts.
7.46 Fourth alarm, Hudson and Franklin Sts.
7.54 First alarm, Bowery and Hester Sts.
8.17 Automatic, Mercer and Houston Sts.
8.19 Second alarm, Bowery and Hester Sts.
8.29 Second alarm, Mercer and Houston Sts.
8.32 Third alarm, Bowery and Hester Sts.
8.40 Third alarm, Mercer and Houston Sts.
8.43 Fourth alarm, Mercer and Houston Sts.
8.45 Fifth alarm, Mercer and Houston Sts.

2 In due time seven pumps were put into operation, with a discharge which reached at times over 35,000 gal. per min., and it was estimated that over 52 fire streams were in service at the same time. Each pump responded instantly and remained in service until ordered shut down. The pressure was ordered gradually increased from 125 lb. to 230 lb., where it was maintained throughout the greater part of the time that the fires raged. The operating force at each pumping station consisted of but one engineman, one oiler, one telephone operator and one laborer.

PROF. GEORGE F. SEVER. A question was asked as to the feasibility of using the storage battery capacity to invert the rotaries and provide alternating current, to be spread through the alternating-current system to the sub-stations, and from those to provide alternating current to the pumping stations. In our preliminary investigation, if I recall the facts correctly, we were assured that this could be done; giving us another feature of reliability in the operation of the system. If the Waterside station should go out of business, we could still get current from the sub-station.

A. C. PAULSMEIER.[1] While the reasons given in the paper for the selection of electric-driven turbine pumps do not coincide with the conclusions as to reliability that have been reached in the West, there can be no question about the careful study given by the engineers who planned the high-pressure fire system described.

[1] Chief Engineer, Byron Jackson Iron Works, San Francisco, Cal.

2 The pumps show a remarkable efficiency, and one of the principal points that should commend them to those interested is their great flexibility as to capacity, a characteristic that every fire pump should possess.

3 The eight fire pumps now being built for the City of San Francisco are of a combined capacity of 216,000 gal. per min., under a working pressure of 300 lb. Each of these pumps is driven by a 750-h.p. Curtis steam turbine, operating at a normal speed of 1800 r.p.m.

4 In addition there are now being completed four fire pumps for the boats Dennis Sullivan and David Scannel, of an aggregate capacity of 9000 gal. per min. under 300 lb. working pressure, or 18,000 gal. per min. under 150 lb. working pressure, the pumps being so arranged that they work either in series or in parallel. The pumps have all been subjected to 24-hr. tests, and while the data on these tests are not sufficiently complete for publication, they show that the pumps are not as flexible as to capacity, or are not as capable of pumping an excess quantity of water, as are the Manhattan pumps. The reason for this is that the impellers in the San Francisco pumps are only 13⅝ in. in diameter, while the inlet to the impellers is less than 10 in. in diameter, this opening being further restricted by the pump shaft, so that it is impossible to obtain much excess water, no matter how much below the normal the discharge pressure is carried.

5 In the station pumps now being built the velocities at the entrance to the impellers have been somewhat decreased, although it is impossible to make anything like the excess capacity shown by the Manhattan pumps, which have impellers of such a size that the inlets may be made anything consistent with good practice.

Prof. W. B. Gregory. It is gratifying to know that efficiencies ranging from 70 to 80 per cent may be obtained with well-designed five-stage turbine pumps. The high-pressure fire-service pumps in New York represent one extreme of conditions, while at the other extreme is the centrifugal pump used in the rice irrigation territory of Louisiana and Texas for raising large quantities of water through comparatively small lifts.

2 The improvement in design of pumps of the latter class in the last ten years, and especially in the last five years, has made it possible to specify an efficiency of 75 per cent, even with heads as low as 10 ft. Purchasers of pumping plants in this section are no

longer satisfied with pumping outfits having efficiencies ranging from 50 to 60 per cent.

3 As examples of the results obtained with pumps of the class that deals with large volumes of water, the tables are quoted from recent acceptance tests conducted by the writer, of pumping plants used for rice irrigation.

TABLE 1 ACCEPTANCE TESTS

TANDEM-COMPOUND CONDENSING ENGINES, DIRECT-CONNECTED

Cane and Rice Belt Irrigating Company, Fulshear, Texas, August 12 and 14, 1908

WORTHINGTON PUMPS	FIRST LIFT	SECOND LIFT
Size of pump (diameter discharge pipe), in	45	45
Water pumped, gal. per min	47,620	46,430
Head on pump, ft	33.90	13.95
Efficiency of engine and pump, %	69.5	73.6
Efficiency of pump (engine 93 %)	74.7	79.2

CROSS-COMPOUND CONDENSING CORLISS ENGINE, DIRECT-CONNECTED

Sabine Canal Company, Vinton, La., May 22, 1909

WORTHINGTON PUMP	
Size of pump (diameter discharge pipe), in	45
Water pumped, gal. per min	44.010
Head on pump, ft	23.26
Efficiency of engine and pump, %	69.5
Efficiency of pump (engine 90 %)	77.3

TANDEM-COMPOUND CONDENSING CORLISS ENGINE, DIRECT-CONNECTED

Second Lift, Neches Canal, July 16, 1909

MORRIS MACHINE WORKS PUMP	
Size of pump (diameter of discharge pipe), in	48
Water pumped, gal. per min	60,300
Head on pump, ft	10.12
Efficiency of engine and pump (maximum), %	69.9
Efficiency of pump (engine efficiency 93.2 % max.)	75

CHARLES B. REARICK. Electrically driven fire pumping-stations for large cities are dependent upon current from an outside source, usually a large central power plant. It would seem quite practicable in many cases to locate new fire pumping stations adjacent to some large power plant having considerable boiler capacity. In such cases it would be possible to drive the centrifugal or turbine pumps with steam turbines, and thus eliminate the necessity of large over-

load capacity in electric generating units for the central station, and also the liability of derangement of the lines between the power stations and the pumping stations. The charge for standby service per annum should be less than for similar electric service.

2 The steam turbines have the advantage of being operative at any speed, and in this manner will maintain in the discharge mains any pressure desired. Furthermore, automatic regulating valves can be used in connection with the turbine to maintain constant pressure irrespective of demand or flow.

3 It is probable that the cost of installation would be less than for electric-driven units. The turbine could run non-condensing, as the question of steam consumption is of small moment for fire service.

HENRY E. LONGWELL. The last paragraph of the paper furnishes a striking illustration of how purely academic is the ordinary official efficiency test, and of how little value as a basis on which to predicate the results that may be expected when the plant is operated under service conditions.

2 This paragraph gives general figures on the performance of the pumps during the fire run. There were 14,095,000 gal. pumped, with a current consumption of 81,450 kw-hr. The average net pressure against which the pumps operated is not stated, but assuming it was 300 lb. per sq. in., the pump efficiency, after allowing for the losses in the motor, would be only 40 per cent. However, we know that for part of the time the pressure did not exceed 225 lb., or, considering the pressure in the suction mains, about 200 lb. net. If the entire quantity of water had been pumped against this lower pressure, the efficiency would be well under 30 per cent. It is therefore perhaps fair to assume that the actual average efficiency was not far from 35 or 36 per cent, or say, in round numbers, only one-half that shown on the official test, when the load and other conditions of operation were more favorable.

W. M. FLEMING. With the rapidly increasing size and height of office buildings, the annual fire loss in the business districts of the cities of the United States is increasing to an alarming extent. The installation of these tremendously effective fire-fighting systems has already proved of definite value in the reduction of city fire losses, and consequently of insurance costs.

2 What was probably the pioneer large and independent so-called high-pressure fire system in this country was installed at

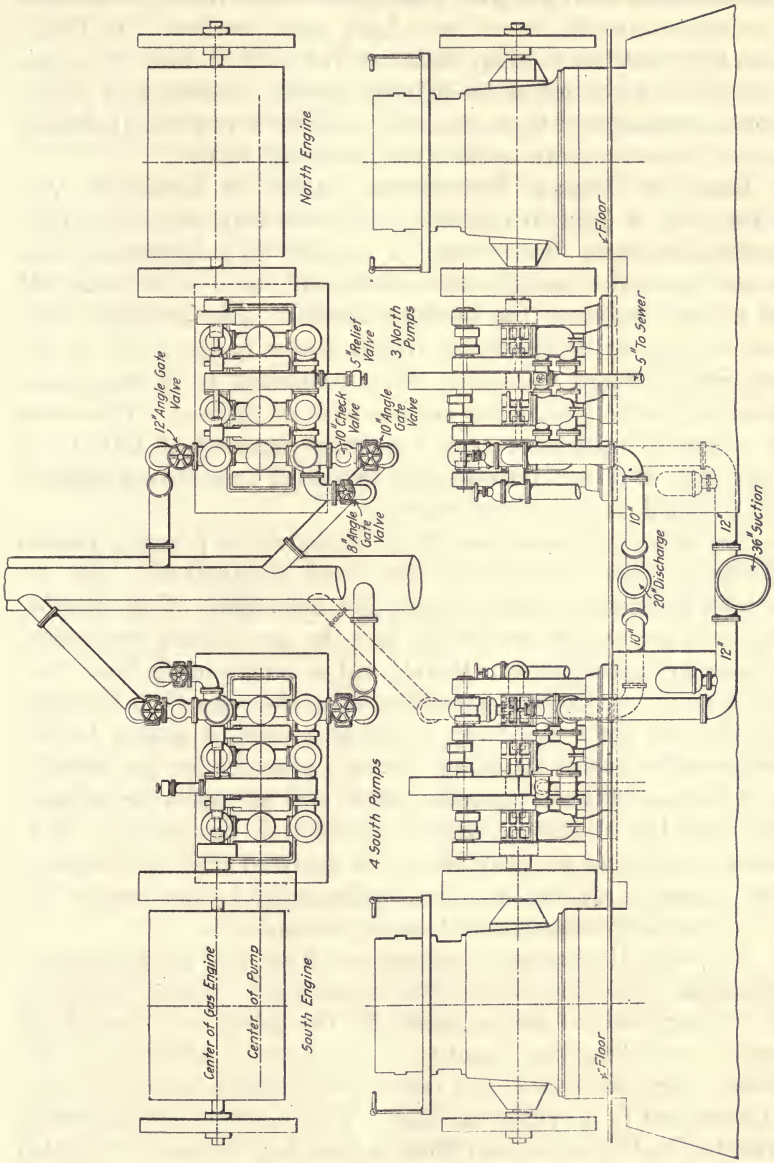


FIG. 1 GENERAL ARRANGEMENT OF THE PHILADELPHIA HIGH-PRESSURE FIRE-PUMPING STATION

Philadelphia in 1903-1904. This plant differs in almost every important detail from the New York system more recently installed; yet the general results in each case have been excellent. In Philadelphia the plant has so many times proved of great value in actual service that a much larger fire-fighting system, consisting of pumping units identical with those originally selected, is now being installed to protect what is known as the Kensington mill district.

3 From the original Philadelphia station at Delaware Ave. and Race St., a location unlikely to be seriously injured by conflagration, Delaware River water is supplied to independent high-pressure fire-service mains which effectually cover more than 425 acres at the center of the business district. The pumping units consist of vertical double-acting triplex power pumps built by the Deane Steam Pump Company, direct-connected to Westinghouse vertical 3-cylinder 4-cycle gas engines each of 280 h.p. The seven large pumping units have each a nominal capacity of 1200 U. S. gal. per min., at 300-lb. pressure, and two small units have a capacity of 350 U. S. gal. at the same pressure.

4 The general arrangement of the Philadelphia pumping station is similar to that of the large New York installations (Fig. 1). Two rows of pumping units occupy the main floor of the station. The pumps are nearest the center, and the gas engines are located in the same relative positions thereto as the motors in the New York pump houses. A platform extending along the sides of the building, about ten feet above the floor, serves as a working gallery for the operation of the engine throttles. Space is provided for the installation of three additional pumping units, and all mains are proportioned with the ultimate probable capacity of the plant in view. Suitable connections are provided to the mains so that the capacity of the pumping station may be supplemented by the use of the city's powerful fire boats, should occasion require.

5 The internal-combustion engines are of the well-known standard Westinghouse type and require little explanation. Speed regulation with varying loads is accomplished by the action of a centrifugal governor controlling the quantity of combustible admitted to the cylinders. Ignition is by a very neat type of make-and-break mechanism contained in a cylindrical plug. Two independent igniters are provided in each cylinder, and three independent sources of ignition current are available at all times. The engines are started by the use of compressed air, which is admitted to one of the cylinders at the proper time to secure rotation in the direction required until the

regular cycle of operation is established. The pumps are started under no-load.

6 The pumps are of the vertical, double-acting piston, triplex power type, requiring comparatively small floor space and giving a rate of discharge so smooth and uniform as to make imperceptible at the hose nozzles any pulsation in pressure.

7 In Fig. 2 is a sectional view of one of the pumps, indicating quite clearly the extreme simplicity and accessibility of the machine, and its general construction. All valves are of the poppet type, readily accessible through handhole openings. Valve areas and waterways naturally are comparatively large, so that friction losses

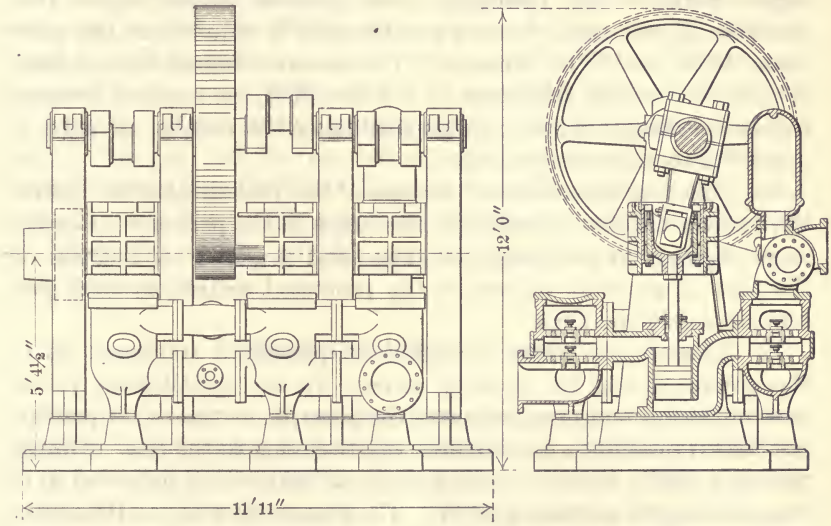


FIG. 2 SIDE AND SECTIONAL END ELEVATION OF TRIPLEX PUMPS FOR THE PHILADELPHIA HIGH-PRESSURE FIRE-PUMPING STATION

are reduced to a minimum. The water ends are thoroughly brass-fitted in order that the pumps may be readily started after a long period of disuse.

8 There is a connection through a 12-in. check valve, from the city mains to the high-pressure system, so that the mains and pumps are constantly primed with a pressure of 60 lb. and are ready for service at all times. A complete system of fire-alarm boxes and telephones, with underground wires, permits direct communication between the vicinity of any fire and the pumping station. On the sounding of the alarm, the station force, consisting of an engineer and his assistant, can bring the total plant of seven large units

into service in seven minutes, and have repeatedly done so. Working pressure is invariably available at the hydrants one minute from the time of the alarm. Such a result would be impossible with ordinary movable apparatus.

9 The pumping units are started up under no-load, by the use of a motor-driven by-pass valve, through which the pump discharges into an overflow, until the normal cycle of operations has been set up in the gas engine, when the switch is closed, causing the by-pass valve to close and the discharge to be directed into the fire mains.

10 Experience has indicated that the maximum pressure of 300 lb. is required only for the most extensive fires, and for fires in the higher parts of tall buildings. The pressure records show that probably 75 per cent of the water pumped is required at not more than 150 lb. to 175 lb. pressure. The pressure desired in each case, is dictated over the telephone by the fire chief, the required pressure regulation being obtained by proportioning the number of units in operation to the requirements.

11 The practical results of the use of the Philadelphia fire system have been: material reduction in fire losses in the protected district, large decrease in fire insurance rates, and a greater willingness on the part of property owners in the protected section to erect pretentious office buildings.

12 Though the writer is unable to present a statement as to the annual saving to property owners by the installation, yet in view of the low cost of operation of the plant, there can be no question but that it presents a considerable yearly saving to the city. During the year 1907, which is perhaps typical, water was delivered to 16 fires, the longest one lasting 44 hr. The plant responded to 116 alarms at which no service was required. The operating expenses for the year were as follows:

Gas, 839,488 cu. ft. at $1.00	$839.49
Electric lighting	343.99
Electric power	7.98
65 tons pea coal at $3.50	227.50
Supplies furnished the pumping station for the entire year 1907	1,500.00
Total fixed charges for 1907	$2,918.96

SUMMARY

Salaries (Total for entire staff)	$8,389.72
Total cost materials	2,918.96
Total operating expenses	$11,308.68
Total daily maintenance charge, salaries and operation	$31.12

13 No mechanical defects have yet developed in either engines or pumps, and practically the only replacements have been a few rubber valves for the pumps and ignition details for the engines.

14 While no definite comparison can be made between the small plant in Philadelphia and the comparatively large plants in New York, which have not yet been in operation for an appreciable length of time, the operating expenses of the Philadelphia plant seem likely to prove much less for a given quantity of service. This is largely due to the so-called "readiness-to-serve" charge made by the company furnishing power to the New York plants. To this charge must, of course, be added the cost of the current actually consumed.

15 Unfortunately no mechanical efficiency test has ever been made on any of the Philadelphia pumping units. Judging from tests of similar machinery, an efficiency of 80 to 85 per cent is to be expected from pumps of this character operating against 150 to 200 lb. pressure. If this is the case, knowing that 75 to 80 per cent of the water to be used will be required at pressures not to exceed 175 lb., it would seem that the plant efficiency in Philadelphia would prove greater than in New York, where we understand that the water must be delivered through reducing valves from 300 lb. to any lower pressure required.

DISCUSSION AT ST. LOUIS

HORACE S. BAKER[1] presented some very complete notes on the proposed high-pressure system for Chicago, an abstract of which is given herewith. After telling of that city's need of a high-pressure system, Mr. Baker illustrated the effect of such an installation on insurance rates by citing the reductions brought about in other cities, details of which are given in Table 1, herewith.

2 The costs of maintaining and operating the proposed system for Chicago should not be more than the following figures, and probably much less:

Operating costs of three pumping stations, including interest and depreciation	$180,000
Interest on cost of distribution system, 4 per cent of $3,000,000	120,000
Depreciation of distribution system, 2 per cent of $3,000,000	60,000
Maintenance of distribution system	50,000
	$410,000

[1] Engineer, Department of Public Works, Chicago.

TABLE 1 HIGH-PRESSURE FIRE SERVICE IN THE UNITED STATES

System	Estimated Population	Date of Installation	Source of Pressure	Gal. Per Min.	Max. Pressure, Lbs.	Lineal Ft. of Mains	Sizes of Mains, Ins.	No. of Hydrants	Total Cost of System	No. of Acres	Cost Per Acre	Connection With Building	Effect on Insurance Rates
Atlantic City	40,000	Proposed	1 Station Elec. Turb. Pumps	7,000	225	38,590	8–14	82	$187,272	306	.$612		
Baltimore	575,000	Proposed	Pump. Sta.	....	....	75,900	10–20	.....	*397,999	360	.....	Standpipes on Buildings	
Boston	620,000	1898	Fire Boat	6,000	200	4,700	12	14	30,080	65	463		
Brooklyn	1,400,000	1906	2 Pump. Sta. Elec. Turb. Pumps	32,000	300	.....	8–20	....	1,384,500	1420	975		
Buffalo	420,000	1897	3 Fire Boats	.....	300	12,756	12	.....	.....	.....	.....	.....	Reduction of $0.30 per $1000
Chicago	2,229,000	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Cleveland	480,000	Constructing	2 Fire Boats to have Pump. Sta.	10,000	300	32,524	8–20	96	*170,000	338	.....	May have Connection with Auto. Sprinklers.	Reduction of $0.80 per $1000 Prop.
Coney Island	.....	1905–6	1 Station Gas Triplex Pumps	3,600	150	.....	8–16	.....	90,000	147	612	.....	Reduction of 25 %
Detroit	380,000	1893	2 Fire Boats	10,000	210	25,831	8–10	95	.....	356	135	.....	Probably has Prevented Increase
Fitchburg	33,000	.....	Gravity†	.....	180	28,250	8–16	.....	50,000	346	144	Boiler Feed, Elevators and Sprinklers	Prevented Increase
Hartford	98,000	Proposed	1 Station	10,000	300	53,430	8–24	198	796,277	731.3	1089	No open Connection	.....
Lawrence	75,000	1906	Gravity †	.....	134	10,200	10–12	39	.....	120	.....	.....	No change
Milwaukee	340,000	1889	3 Fire Boats	15,000	250	45,717	6–12	183	.....	630	.....	.....	10% Reduction

Newark.....	290,000	1905	Gravity	3,500	165	15,000	20–30	52	135,000	303	446	Some Connection	10% Reduction
New York..... (Manhattan)	2,100,000	1908	2 Stations Elec. Turb. Pumps	30,000	300	.....	12–24	1200	3,950,400	1430	2763	Water Curtains Provided for	No change
Philadelphia.....	1,500,000	1903	1 Station Gas Triplex Pumps	9,100	300	35,300	8–16	166	700,000	512	1367	None on or in Buildings	Penalty of 25% Removed‡
Providence.....	200,000	1897	Gravity †	3,472	116	29,000	12–24	89	143,136	358	400	5 Automatic Sprinklers	No change
Rochester.....	185,000	1874	2 Stations Elec. Turb. Pump, Steam Turb. Turb. Pump	9,000	140	102,960	4–20	.....	.....	.....	.....	Some Connection	Graded Reduction
Toronto.....	215,000	Constructing	1 Station Elec. Turb. Pump	14,000	300	40,000	8–20	.....	500,000	287	1742	Considering Connection	Uncertain
Winnipeg.....	110,000	1908	1 Station Gas Producer Gas Eng., Tripl. Pumps	10,800	300	15,840	8–20	.....	650,000	275	2364	Connection with Auto. Sprinkler	Uncertain
Worcester.....	138,000	.....	Gravity†	.....	165	100,320	8–30	.....	.....	1380	.....	Elevators	No change

* Exclusive of pumping station and equipment.

† System consists of extension of pipes from high service into district covered by low service.

‡ Board of Fire Underwriters have voted to reduce rates to the amount of 10 cents per $1000 = a total of $40,000, if extensions costing $150,000 are made to the system.

NOTE:—This table is taken from the report on the proposed high-pressure fire system for Hartford, Conn.

TABLE 2 COST DATA FOR VARIOUS TYPES OF APPARATUS

NAME OF SYSTEM	TYPE	PRESSURE AND CAPACITY PER MIN.	TOTAL COST EXCEPT DISTRIBUTION SYSTEM	ANNUAL OPERATING EXPENSE OF PUMPING STATIONS[1]	COST PER 1000 GAL. PER MIN. CAPACITY[1]	ANNUAL OPERATING COST PER 1000 GAL. PER MIN. CAPACITY[1]
Manhattan	Electric Centrifugal Pumps	300 lb. 30,000 gal.	$670,000	$139,250	$22,333	$4,642
Coney Island	Gas Engines Triplex Pumps	150 lb. 4,500 gal.	47,000	14,186	10,444	3,152
Philadelphia	Gas Engines Triplex Pumps	300 lb. 9,100 gal.	260,000	11,978	28,571	1,316
San Francisco Estimate 1	Steam Turbines Centrifugal Pumps and boiler Plant Oil Fuel	300 lb. 20,000 gal.	622,228	34,630	31,111	1,732
San Francisco Estimate 2	Gasolene Engines Turbine Pumps Rope Drive	300 lb. 20,000 gal.	737,848	30,595	36,892	1,529
Hartford Estimate 1	Steam Turbines Centrifugal Pumps Coal Fuel	300 lb. 12,600 gal.	257,620	45,320	20,466	3,597
Hartford Estimate 2	Gas Engines Triplex Pumps	300 lb. 12,600 gal.	377,905	8,648	29,992	686
Chicago, Estimate	Steam Turbines Centrifugal Pumps	250 lb. 10,000 gal.	263,005	37,400	26,300	3,740
Chicago, Estimate	Gas Engines Triplex Pumps	250 lb. 10,000 gal.	248,112	24,626	24,811	2,463
Chicago, Estimate	Electric Motors Centrifugal Pumps	250 lb. 10,000 gal.	122,882	57,700	12,288	5,770

[1] Exclusive of Interest and Depreciation.

3 In the light of current practice as shown in Table 2, it seems advisable to consider and estimate on the following types of pumping stations:

a Steam turbines and centrifugal pumps.

b Electric motors and centrifugal pumps.

c Gas engines and triplex pumps.

TABLE 3 STEAM TURBINE PUMPING STATION

Approximate Estimate of Cost

Capacity 10,000 gal. per min.; pressure 250 lb per sq. in.

1	Excavation:		
	Pump pit	2300 cu. yd.	
	Boiler room	3865 " "	
	Stack	565 " "	
	Conveyor tunnel	70 " "	
		6800 cu. yd. at $1	$6,800
2	Concrete:		
	Retaining walls for pump pit	616 cu. yd.	
	Boiler room foundations	453 " "	
	Stack foundations	430 " "	
	Pump house foundation	101 " "	
		1600 cu. yd. at $7	11,200
3	Building:		
	Pump room, 60 ft. by 54 ft. = 3240 sq. ft.		
	Boiler room, 78 ft. by 84 ft. = 6552 " "		
		9792 sq. ft.	
	Assume 10,000 sq. ft, by 30 ft. = 300,000 cu. ft. at 15 cents		45,000
4	Foundations for pumps and turbines, 150 cu. yd. at $10		1,500
5	Four 2500-gal. centrifugal pumps at $5,000		20,000
6	Four 600-h.p. steam turbines at $12,000		48,000
7	Boilers, 2400 h.p. at $15		36,000
8	Chain grates, hoppers, conveyors, etc		15,000
9	Stack		8,000
10	Suction piping from city main and tunnel		6,500
11	Discharge piping		5,000
12	Steam piping		7,500
13	Condenser		6,200
14	Boiler auxiliaries, heater, purifier, pumps, etc		9,000
15	Two 20-in. venturi meters and recorders		3,000
			$228,700
	Add 15 per cent		34,305
			$263,005

Approximate Estimate of Operating Expense

1	Interest, 4 per cent of $263,005	$10,520
2	Depreciation, 4 per cent of $263,005	10,520
3	Coal:	
	200 hr., 5 tons at $2.50 }	
	8560 hr., ½ ton at 2.50 }	13,200
4	Oil, waste and supplies	1,500
5	Repairs	2,500
6	Labor:	
	Men, cost per annum three 8-hr. shifts:	
	1 engineer $6600	
	1 oiler 4500	
	1 fireman 3000	
	2 coal passers 5400	
	1 janitor 700	
		20,200
	Total	$58,440

TABLE 4 GAS-ENGINE PUMPING STATION

Approximate Estimate of Cost

Capacity 10,000 gal. per min.; pressure 250 lbs.

1	Excavation:		
	Retaining wall	68,400 cu ft.	
	Main pit	58,089 " "	
	Engine foundations	5,096 " "	
	Pump foundations	7,056 " "	
	Tunnel	5,496 " "	
		144,137 cu. ft.	
		= 5,339 cu. yd. at $1	$5,339
2	Concrete:		
	Retaining wall	11,520 cu. ft.	
	Retaining wall footing	23,040 " "	
		34,560 cu. ft.	
		= 1,280 cu. yd. at $7	8,960
3	Building: 82 ft. by 79 ft. by 30 ft. = 19,430 cu. ft. at 15 cents		29,151
4	Foundations for pumps and engines, 450 cu. yd. at $10		4,500
5	Seven 1500-gal. triplex pumps, for 250 lb. pressure at $8900		62,300
6	Seven 300-h.p. gas engines at $10,000		70,000
7	Freight and erection		7,000
8	Suction pipes from city main and tunnel		6,500
9	Water discharge pipes		5,000
10	Gas connections		8,000
11	Air compressor plant		2,500
12	Gasolene tanks and piping		3,500
13	Two 20-in. venturi meters and recorders		3,000
			$215,750
	Add 15 per cent		32,362
			$248,112

Estimate of Operating Expense

1	Interest, 4 per cent on $248,112		$9,924
2	Depreciation, 4 per cent on $248,112		9,924
3	Gas: 200 hr. at 18 cu. ft. per h.p. at $0.85 per M		6,426
4	Labor: 3 engineers at $2200 =	$6600	
	6 asst. engrs. at 1500 =	9000	
	1 janitor at	600	16,200
5	Oil, waste and supplies		1,000
6	Repairs		1,000
	Total		$44,474

TABLE 5 ELECTRIC PUMPING STATION

Approximate Estimate of Cost

Capacity 10,000 gal. per min.; pressure 250 lb. per sq. in.

1	Excavation:	
	Pump pit.....63,936 cu. ft.	
	Retaining wall footings..... 8,640 " "	
	Pump foundations..... 2,048 " "	
	Building wall..... 1,692 " "	
	76,316 " " = 2,826 cu. yd. at $1	$2,826
2	Concrete:	
	Wall of pump pit..... 15,264 cu. ft.	
	Footings..... 7,892 " "	
	Bldg. foundation wall..... 920 " "	
	Bldg. foundation footings..... 329 " "	
	24,405 cu. ft. = 904 cu. yd. at $7.....	6,328
3	Building:	
	Pump room, 36 ft. by 56 ft. = 2016 sq. ft.	
	Switch room, 16 ft. by 56 ft. = 896 sq. ft.	
	2912 or say 3000 sq. ft.	
	3000 sq. ft. by 30 ft. = 90,000 cu. ft. at 15 cents	13,500
4	Foundations for pumps and motors, 150 cu. yd. at $10.....	1,500
5	Four 2500-gal. centrifugal pumps at $5000.....	20,000
6	Four 600-h.p. 3-phase induction motors at $10,800.....	43,200
7	Suction piping from city main and tunnel.....	6,500
8	Discharge piping and valves in station.....	5,000
9	Switchboard and wiring in station.....	5,000
10	Two 20-in. Venturi meters and recorders.....	3,000
		$106,854
	Add 15 per cent.....	16,028
	Total.....	$122,882

Approximate Estimate of Operating Expense

1	Interest, 4 per cent of $122,882.....	$4,915
2	Depreciation, 4.3 per cent of $122,882.....	5,284
3	Power bill:	
	Ready-to-serve charge, $25 per kw. = $37,500	
	$0.005 per kw. per hr., 200 hr. of full load $1,500.....	39,000
4	Labor, 3 shifts:	
	3 engineers $6600	
	6 asst. edgineers 9000	
	1 janitor 600	16,200
5	Miscellaneous: oil, supplies, etc.....	1,500
6	Repairs.....	1,000
		$67,899

TABLE 6 ESTIMATED COST OF PROPOSED CHICAGO SYSTEM

MAINS, VALVES AND HYDRANTS

DISTRICT NO.	COST
1	$477,508
2	329,321
3	152,018
4	128,457
5	109,178
6	314,569
7	82,791
8	178,420
9	146,432
10	118,916
11	113,268
12	85,852
13	75,918
14	175,811
Total	$2,488,459
Engineering and contingencies	373,269
	$2,861,728
4 stations at $250,000 =	1,000,000
	$3,861,728

No allowance made for land.

River crossings are assumed to be made as follows: (*a*) North branch in present Grand Ave. water pipe tunnel; (*b*) Main River in proposed LaSalle St. water pipe tunnel, to be built by Chicago Railways Company; (*c*) South branch in present Harrison St. water pipe tunnel.

4 For the purpose of estimate it seems proper to assume a station of a capacity of 10,000 gal. per min. against 250-lb. pressure, the working pressure to be probably 150 to 200 lb. To avoid the crippling of a station by the shutdown of any unit it seems advisable to consider units of 2500 gal.

5 In discussing the various types of installations proposed, Mr. Baker cited the advantages of each type. The direct-acting duplex pumps are rugged and ready for immediate service, but their steam consumption is large. The independent boiler plant necessary, moreover, would be costly to build and to operate.

6 The gas-engine station has the advantage of lower first cost, and no cost for power when not in operation. Though failure of the gas supply is unlikely, gasolene could be used with a change of

adjustment, or by running normally on illuminating gas with low compression, which would be somewhat uneconomical. A gas-producer plant might be installed, though this is somewhat open to the same objection as the boiler plant.

7 Though electric motors are supplied from an outside sources the large number of generating stations and feeders makes the electric, supply as reliable as the gas supply. The first cost and the operating expense of an electric station are low, though the standby charge is high.

8 Connecting the system to stand pipes and to the sprinkler systems in buildings had been recommended in Chicago and is the practice in Winnipeg, Man., and Providence, R. I., and also with the gravity system in Newark, N. J., Worcester and Fitchburg, Mass. The fire systems of New York City and Philadelphia are not connected in this way. The objection to these connections is that great loss of water might result from broken pipes in the buildings. This could be avoided, however, by placing a controlling valve in a brick chamber outside the curb.

Edward E. Wall[1] outlined the proposed fire system for St. Louis, which contemplates the installation of six or eight 5-stage centrifugal pumps, electrically driven, at a station on Chestnut St., from which the fire service mains will radiate north, south and west. The supply for these pumps will be taken from the distribution system, a 36-in. main being laid directly from the Bissell's Point pumping station to the Chestnut St. station, and connected to the present distribution system by a number of by-passes. Connections will also be made between two 20-in. mains on Fourth and Seventh Sts., to the supply for the pumps, so that in case of failure of the 36-in. main, the pumps may be supplied from this source.

2 It would be practicable to draw the fire pump supply directly from the Mississippi River by building an intake, but this would probably cost more than the laying of the 36-in. main, and would necessitate a charter from the Government. It would also raise the question of obstructing navigation, since it would be necessary to carry the construction well out into the channel, to insure an ample supply of water. Supply from the river direct would also preclude ali connection with the distribution system, as it would be unwise to risk the contamination of the city's water supply by river water.

[1] Asst. Water Commissioner, St. Louis.

3 The pumping capacity of the station at Bissell's Point will be over 100,000,000 gal. of water every twenty-four hours, which is more than twice the amount ordinarily consumed; the excess being sufficient to supply more than 30 fire-streams through 3-in. hose continuously, assuming 300-lb. pressure at the fire pumps.

4 The 5-stage centrifugal pumps proposed for the Chestnut St. station will have a capacity of 150,000 gal. per hr. each, against a pressure of 300 lb. per sq. in. It is proposed to connect the station with the power plants of the Union Electric Light and Power Company and the United Railways, so that two sources for power will be available.

5 The three discharge mains from these pumps will be 24 in. in diameter, the district supplied by them to be gridironed by a system of 12-in. mains laid on the enclosed streets and occasionally connected, at crossings only, by by-passes, that the breakdown of one main may not necessitate the cutting out of any other line. The pipe used will be cast iron, extra heavy, with bell and spigot joints, double-grooved. All fire-hydrant leads will be 8 in. in diameter.

6 The system will be under the ordinary distribution pressure when the fire pumps are not in use, so that for small fires the hydrants will be available for use; when the fire pressure is put on the system, the check valves on the by-passes will prevent additional pressure from coming on the distribution system.

7 While the arrangement of machinery for the pumping station, and the details of operation, have not been definitely decided upon, it is possible that gas engines may be used instead of electric motors. The questions of automatically starting and stopping the pumps, maintaining the pressure during a fire, and the general details of operation of the station, as well as the minor points of weight of pipe, design of hydrants, etc., have all to be worked out. It is estimated that the cost of this system will approximate $3,000,000.

H. C. Henley[1], speaking on the advantages of high-pressure fire systems, said that they were chiefly valuable for the numerous powerful streams which can be quickly brought into service and concentrated to advantage. For the prevention of conflagrations and for keeping serious fires from spreading, more powerful streams are needed than can be supplied by portable fire engines without considerable delay. To obtain such streams from fire engines, it is necessary to "siamese" two or more lines into one nozzle, requiring considerable time; and

[1] Chief Inspector, St. Louis Fire Prevention Bureau.

if a change in the location of engines becomes necessary, considerable time is again lost in re-assembling the hose lines.

2 The high-pressure system permits the use of hose of large diameter—3 in. and 3½ in.—and direct connection to hydrants furnishes a supply to nozzles of large area, without the necessity of siamesing two or more hose lines. The 2-in. nozzle is best adapted for use with high-pressure systems, this nozzle, under 75 lb. nozzle pressure, discharging approximately 1000 gal. per min. A nozzle of this area provides very effective service, as the loss of pressure, due to friction in fire hose, decreases as the area of the hose is increased. The data given in the table are derived from experiments by John R. Freeman, and show the pressure required at the hydrant in

PRESSURE REQUIRED AT HYDRANT TO OVERCOME FRICTION LOSS

Hose Diameter		2½ IN.		3 IN.		3½ IN.	
Hose lines		Single	Siamesed	Single	Siamesed	Single	Siamesed
Smooth bore nozzle		1¼ in.	2 in.	1¼in.	2 in.	1¼ in.	2 in.
Length of hose line, ft.....	100	121	139	92	101	84.5	88
	150	139	170	99.5	113	87.5	93.5
	200	158	201	107	125	91	99
	250	176.5	232	114.5	137	94.5	104.o
	300	195	263	122	149	98	110
	400	232	325	137	173	105	121

For the 2-in. nozzle it is assumed that two hose lines of the length given are siamesed together.

hose streams of various lengths, to overcome friction loss and maintain 75-lb. nozzle pressure, the nozzle being at the same level as the hydrant.

3 High-pressure systems should be considered as auxiliary protection and there should be no attempt at abandonment of engines or other apparatus.

4 Direct connection from a high-pressure system to interior standpipes, sprinkler equipments and open sprinkler systems, should be made through siamese connections and not through direct pipe connection.

5 The inability of portable steam fire engines to furnish a stream efficient to cope with serious fires is made apparent by tests made by the engineers of the National Board of Fire Underwriters. The steam fire engines for test were picked at random from the equipment of many of the best city fire departments in the country.

Number of engines tested	102
Nominal capacity, gal	69,800
Actual capacity, gal	55,900
Percentage of efficiency	80

In many cases the efficiency of individual "steamers" is less than 50 per cent.

EDWARD FLAD. It appears to me that a cast-iron pipe is rather dangerous for high pressure. A cast-iron pipe tested under 300-lb. pressure will often break at 75 lb. A wrought steel pipe is much more reliable, and if properly coated, should last 25 or 30 years under ordinary conditions. If steel pipe is absolutely reliable we can afford to relay it at the end of 25 years rather than to use cast-iron pipe, which is liable to break.

2 In answer to a question by Mr. Flad as to the flexibility of the joint used in Baltimore, Professor Carpenter replied that it is flexible, in the sense that it can be laid at an angle; it is not flexible so far as change of form is concerned.

H. S. BAKER asked what kind of steel pipe would be used in Baltimore, Professor Carpenter answering that it is extra heavy steel welded pipe, $\frac{7}{16}$-in. thick, the ends being expanded into semi-spheres, an 8-in. or 12-in. pipe being expanded just enough to get a ring in it, and the whole bolted on the outside by external bolts, very like a steam pipe.

PROF. H. WADE HIBBARD. It is a fact that a cast-iron water main has been in satisfactory use in city service for twenty years and then a piece has blown out. It seems to me that the use of cast-iron pipe should be prohibited for this special emergency purpose of fire protection on account of its unreliability. In fact, in one of the high-pressure systems using cast-iron mains, leaks have been known to take place and the pumps to run for a considerable interval, some hours, I will say, and the pressure could not be maintained under test, until it was finally discovered that the water had been pouring out into a very large excavation and flooding it, unknown to those operating the pumping station. Steel will show approaching deterioration as cast iron will not.

2 Steel pipe ought to be good for thirty years of service. That period of service should be sufficient, and cities having such pipe

hould then be willing to replace it, having had more reliable protection during that period of years, than cast-iron pipe could possibly give.

H. C. HENLEY asked if there had been any attempt made to prevent the pipes from deteriorating through electrolysis, Professor Carpenter answering that the Baltimore system is a continuous metallic structure, from one end to the other, and he believed would be thoroughly protected from electrolysis; or at least, better than by any other system.

E. E. WALL. It is a fact that actually and not figuratively, steel pipe must be handled with gloves when it is laid, because the coating has to be very carefully preserved and can hardly be repaired if it is broken in handling before the pipe is laid. This is a very serious objection to the laying of steel pipe on account of exposure to corrosion after it is laid.

W. H. REEVES. Owing to the magnitude and prominence of these plants, the pump performances should be of interest to those desiring information on centrifugal and turbine pumping machinery. The foremost consideration in the art of building machinery of this class is accuracy in design. Without accuracy in design it is not possible to secure the maximum efficiencies within reach. A closely designed pump should deliver exactly its contract number of gallons against the contract pumping head, and the capacity should not run over nor under. From a pump builder's point of view the misfortune of falling short of the contract capacity needs no discussion here, but the other misfortune of running over on capacity may not be so clearly understood. One effect of running over is an overload on the motor, engine or steam turbine driving the pump, and another result is that the average efficiency of the equipment in daily operationuis below what it should be, for if it runs over in capacity its maximum efficiency does not occur at its contract capacity.

2 It will be noted that each of these pumps had a contract capacity of 3000 gal. per min., against a total head of 308.66 lb. per sq. in. Table 2 of the paper shows the performances of the five pumps at the South Street pumping station. This table does not show the averages, but it will be found that each pump averaged approximately 3761 gal. per min. against a mean total head of about 313.1 lb. per sq. in. Although the head was about 5 lb. above the contract condition, the pumps exceeded the contract capacity by about 25 per cent. This, no

doubt, caused the motor overload mentioned in Par. 64. The contract conditions implied 540 h.p. actually delivered, and at the guaranteed pump efficiency 770 b.h.p. would be needed. The delivered work under test was 686 h.p., and according to the test efficiency of 72½ per cent, 946 b.h.p. was used, that is, approximately 23 per cent excess motor load.

3 There appear to be no data on tests made on contract conditions. As the pumps were tested at a great excess in capacity it is quite probable that the efficiency would have been lowered several points if the pumps had been throttled to the agreed capacity and head. The tests as per Table 2 show about 686 h. p. delivered and 946 b.h.p., or a pump loss of 260 h.p. For a considerable range it is probably safe to assume this 260 h.p. loss to be fairly constant. Assuming this to be correct and adding this loss to the 540 h.p. delivered represented by the agreed contract conditions, would give 800 b.h.p., thus showing a pump efficiency of but 67½ per cent. If these pumps had been accurately designed, undoubtedly they would have shown as high efficiency under the contract conditions as was obtained with excess capacity condition.

Prof. E. L. Ohle. There seems to be quite a difference in opinion among engineers as to the reasons for the variation in efficiency of the pumps when working singly and in multiple. It seems to me that the reason is the one suggested by Professor Carpenter. It is practically impossible that all should work at the same speed, as they are independently driven. If then the pressure in the main should exceed the pressure which any pump was capable of delivering, the runner of that pump would simply revolve without delivering any water. This seems to be borne out by the experience of one pump builder, as stated by J. J. Brown.

The Author. The discussion of the paper has been so voluminous that there is really but little needed from the author. In most of the discussion additional information of value has been contributed which I am sure will be appreciated by members of the Society.

2. The difficulties in connection with an installation of the kind described in the paper, involving a complete system of piping and hydrants capable of withstanding high pressures, as well as the necessary pumping machinery, are well brought out. I think the general conclusion will be that the piping difficulties to be overcome, especially when cast iron is employed, are very serious and require

special skill and the best of material. Attention has also been called to the fact that the city of Baltimore has adopted a system in which steel pipe is employed in order to overcome the difficulties due to the breakage of cast-iron pipe.

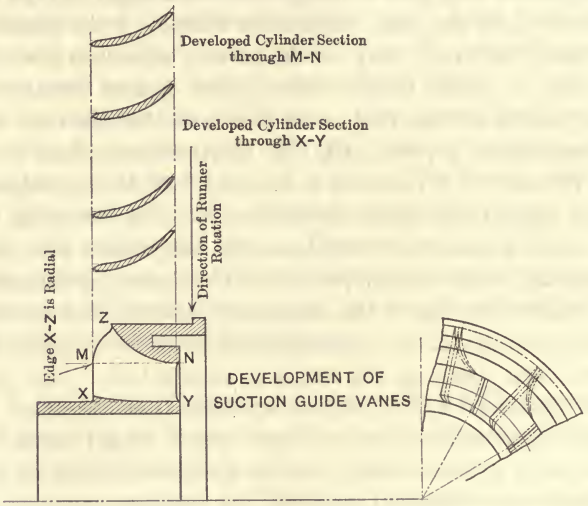
3 The discussion has disclosed the construction of several stations in which the motive power has been obtained from gas engines, and the advantages, disadvantages and expense of such installation.

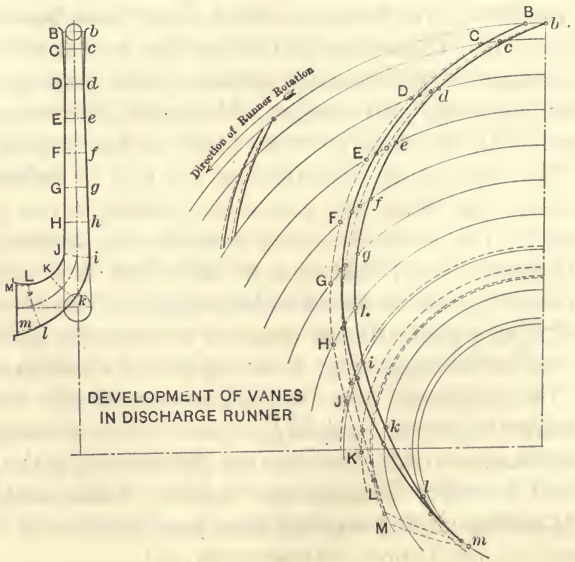
4 It is pointed out that although the centrifugal pumps are capable of operation at the high efficiencies shown by the paper, yet at the lower heads at which they are frequently operated the efficiency would be less. I do not believe there is any serious commercial disadvantage because of that fact, since it is true that the cost of operation of a fire station is principally due to other items than the cost of power. A fire station is required to be, above all things, reliable, and it is of very little importance whether or not the pumping be done under the most economical conditions for the reason that the total cost of pumping is only a small portion of the operating expense.

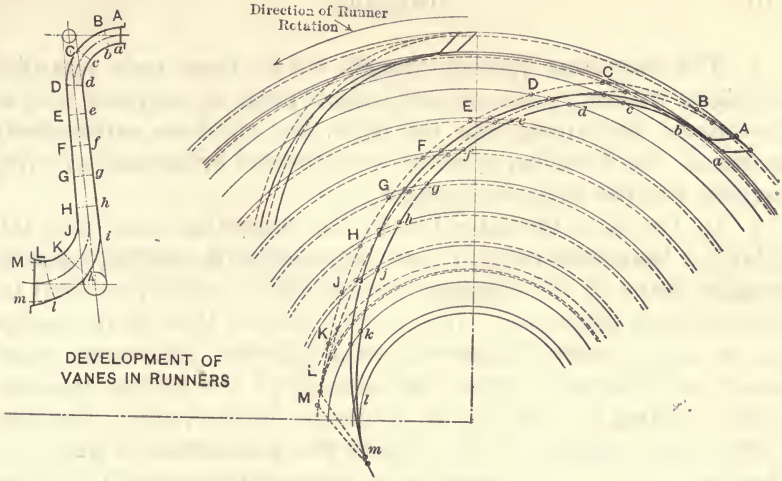
5 It is claimed by one of the discussors that the test should have been made by the city at the exact capacity called for and the efficiency should have been based on the result of such a test. This doubtless would have produced a lower efficiency than that obtained. In the light of the information now at hand, there would have been no injustice in such a requirement, but at the date of making the contract matters were different and such a requirement would have imposed a penalty on the builders, which would have been of no advantage to the city. The reason for that opinion is, that at the time of taking the contract the information regarding multi-stage pumps operating at high heads was quite meagre. Mr. Sando, the designer of the pumps, secured all the data he could both in this country and in Europe. The result of his investigation led him to believe that it was to the advantage of the city and of the builders to put in a pump of such capacity that it would surely meet the requirements in that respect. It was believed that this would result in a considerable increased capacity over contract requirements. The motors were designed with an equally liberal capacity so that the machine was intended, even in the beginning, to be capable of a continuous large overload. The statement that the motors showed any evidence of being overloaded is in error, possibly because a certain remark which I made was misunderstood. It strikes me that the city is the principal gainer by such a system of design and that as a consequence it owns considerable more pumping capacity than was called for in the specifications, and so far as I know, without extra cost.

6 I believe that with the present data it would have been possible to design both pumps and motor to carry 25 per cent less load with the same efficiency as was obtained by the larger pumps and motors. In that case, a test at the specified capacity would have been a fair one.

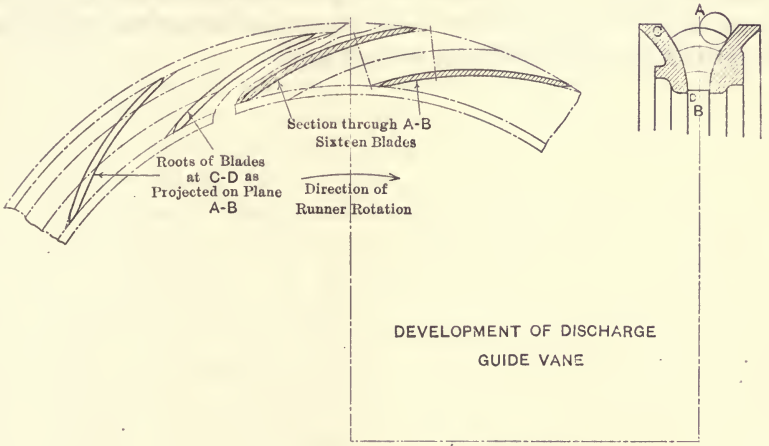
[The following curves show the development of the runners, guide wheels and guide vanes of the pumps installed in the New York high pressure pumping stations.—EDITOR.]



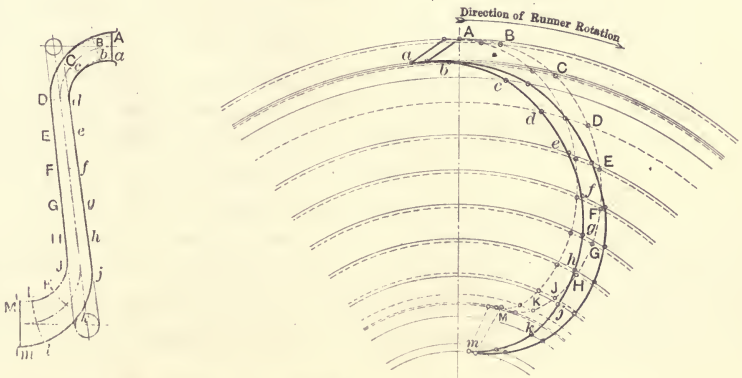




DEVELOPMENT OF VANES IN RUNNERS



DEVELOPMENT OF DISCHARGE GUIDE VANE



DEVELOPMENT OF VANES IN STATIONARY GUIDE WHEELS

7 The interesting question brought out by these tests regarding the higher efficiency obtained with a single pump as compared with all the pumps discharging into the main, has not been satisfactorily answered. Such results, however, seem to have been noted by every engineer who has made similar tests.

8 In Par. 65 of the paper I made one suggestion concerning this point. I have since thought that the variation in construction or in detailed shape of the discharge volume might possibly account for some of these differences. It is hardly possible that all the pumps can be made exactly alike and small inherent differences, which would be obliterated in the operation of all the pumps together, might account for the higher efficiency of the pumps operating singly. As suggested by Mr. White, the measurements were of a character which did not consider the pipe resistances, and the figures given apply to the delivery from the pump before the water was subjected to pipe resistances in any case.

No. 1262

THE BEST FORM OF LONGITUDINAL JOINT FOR BOILERS

By F. W. Dean, Boston, Mass.
Member of the Society

It has been generally accepted in this country for a number of years, that the best form of butted longitudinal riveted joint for boilers is that in which the inside strap is wider than the outside, and which has one or more rows of rivets passing through the shell and the inside strap beyond each edge of the outside strap. The pitch of the first row of outer rivets is double that of the rows that pass through both straps, and if there are other outer rows they may or may not have a still greater pitch.

2 In England, where until comparatively recently boiler construction has been superior to ours, this form of joint appears to receive no recognition. It was first devised, as far as I know, by Dr. E. D. Leavitt, Past-President of the Society, and Edward Kendall, both of Cambridge, Mass., and was first used by Mr. Leavitt in some locomotive type boilers designed by him for the Calumet & Hecla Mining Company. I have a blueprint of this boiler dated 1879. It is, of course, hazardous to state that this joint was never used before and it is quite possible that it was used in England, and discarded and forgotten as poor construction, as I believe it is. It was first used on an American locomotive by the Baldwin Locomotive Works in a consolidation locomotive built by them for the Calumet & Hecla Mining Company, the drawing of this joint having been made by me when I was in Mr. Leavitt's employ.

3 While every boilermaker has for years been familiar with butt joints, this form made slow progress towards adoption in this country. One form of joint used to avoid the butt joint and get something as good, was a lap joint with an inside strap bent at the edge of the lap and riveted on each side of it. This was used on locomotives exclu-

Presented at the Annual Meeting, New York, (December 1909), of The American Society of Mechanical Engineers.

sively, and was of little or no value as it was simply a somewhat elastic bent tie connecting the two parts of the shell plate. Finally, and fortunately, this joint gave way to the butt joint first described.

4 I believe there has been no case of an explosion of a butt-joint boiler; at least one due to rupture of the joint. Recently, however, a boiler at Woonsocket, R. I., narrowly escaped explosion, a longitudinal rupture of the plate on one side of the joint, and within its limits, being discovered while the boiler was subjected to steam pressure. The steam pressure was rapidly reduced and no explosion occurred. An account of this is given in Power, January 26, 1909, and the joint itself is in possession of the boiler-inspection department of the Massachusetts district police at the state house in Boston.

5 It has been growing upon me for some years that a one-sided boiler joint, such as that first described, is poor construction, and may sooner or later cause a crack in the plate. The Woonsocket phenomenon has tended to confirm this opinion. It is evident that unless the outside rivets fill the holes they do very little good, and when they do fill them they form an overhung connection and to some extent possess in themselves the now recognized defect of the lap joint. Moreover the extended inside plate forms a bent connection between the different rivets at different distances from the center line of the joint.

6 In many cases designers have placed the outside rivets at a considerable distance from the edge of the outside strap and this is constantly overdone. It is obvious, on careful thought, that the outside rivets should be as near the edge of the outside strap as practicable, thereby diminishing the bent-tie effect. In order to diminish this effect still further, and also to render the overhung rivets more effective, the inside strap should be thicker than usual, and this feature can hardly be overdone. The inside strap should be at least as thick as the shell plate, and great care should be taken to have the holes match and the rivets fill the holes.

7 When a joint of this kind is tested to destruction in a testing machine, it will be found to fail somewhat in detail, the inside strap bending slightly and the outside rivets being the last to rupture after yielding a little. In a boiler the joint would be weaker than a flat specimen on account of the bent-tie feature. This could be prevented if it were practicable to calk the inside strap, as it would thereby be compelled to maintain the circular form. The theoretical efficiency of this joint is greater than of any other kind, but in practice I believe the efficiency is not realized and the defects that I have described render the joint, in my opinion, undesirable.

8 In order to avoid the defects of the one-sided butt joint, I have adopted and intend to use hereafter, a joint with both straps of the same width, as illustrated in Fig. 1. This has the merit of having all rivets in double shear and the strains all taken care of in the best manner. The efficiency of this joint can hardly be above 84 or 85 per cent while that of the one-sided joint can be theoretically 91 or 92 per cent; but the certainty that the efficiency of the former is realized in practice is ample compensation for the use of slightly thicker plates. The pitch of the outer rows of rivets is rather great, compelling the use of a thick outside strap in order to stand calking and remain steam-tight. I use an equally thick inside strap in order to diminish the bent-tie effect. This effect is small, however, as the

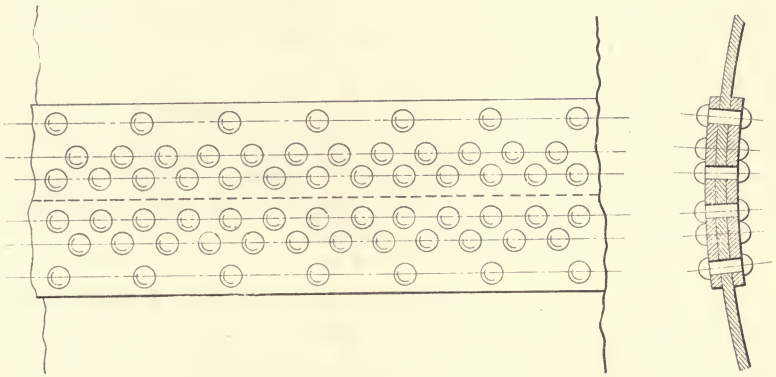


FIG. 1 RECOMMENDED FORM OF LONGITUDINAL JOINT

rivets are all near the center of the joint. It can be eliminated by calking the inside strap, which is practicable with this joint, and is done in the best marine practice. This assumes that the calking is effective and will remain so.

9 While this subject is under consideration, it is well to call attention to the perfection with which the longitudinal joints of boiler-plate cylinders can be welded, a fact which has been demonstrated for many years with corrugated furnaces and more recently with soda digesters. While the joints of corrugated furnaces are in compression those of digesters are in tension, and their proved safety should be sufficient to overcome any timidity concerning the perfection and safety of welded joints. Circumferential joints are not so easily welded as longitudinal, and it is of course of little importance in boilers that they should be welded.

DISCUSSION

REGINALD P. BOLTON. The form of longitudional joint for boilers which Mr. Dean has described as the best is as old as the time of Brunel, and was tested by him in 1838, and again by Longridge in 1857. It is a double-welt triple-riveted joint, omitting alternate rivets in the outer strip, and it has the defect of undue distance for calking between the outer rivets. It is not so good a joint as it would be when the triple riveting is continued, instead of omitting the alternate outer rivet. The other form of joint to which Mr. Dean refers, in which the inside welt was wider than the outside welt, has stood the test of many years usage, and I do not know of any case of failure.

2 In discussing the longitudinal joint, we should not lose sight of the fact that the weak parts of every longitudinal joint are the ends, where the two shell plates unite and the circular seams meet the longitudinal joint. It is there that weakness develops in all joint construction. In explosion cases on which I have been engaged, I have found that trouble has developed at those points, and have noted that ruptures commenced there. Therefore, in dealing with the design of longitudinal joints, the essential feature seems to me to be its character where it meets the circumferential seam.

E. D. MEIER. I think that the value of this joint depends largely on the diameter of the boiler that one has in mind. In a Scotch marine boiler, from 12 to 15 ft. in diameter, the joint would be an excellent one, especially with the scalloped edges mentioned by Mr. Dean[1]. That is a very troublesome thing to do, but in addition to the advantage of the scalloped edge which Mr. Dean cited, there is the further one, that it modifies the tendency, common to such joints, to buckle at the point where the sheets come together. The butt joint is stiffer there than any other part of the shell and with a change in the pressure and temperature the buckling ultimately tends to impair the joint.

2 With a small boiler, 36 in., 42 in., or 48 in. in diameter, the joint is too large a proportion of the total circumference, and this action would become worse. That buckling action is distributed by making the butt plates as thin as possible, and making the inside one longer than the outside one.

3 The welded joint will be an ideal one when we can be sure of

[1] This was referred to by the author in presenting his paper.

a weld that will give 95 per cent efficiency. The difficulty will be to test it. We do know, however, that when we rivet a joint and do it honestly, we have something that can be relied on. Much will depend on how the material is chosen and how the work of laying up and riveting is done. The joint should be made by carefully bending the butt straps at a red heat to the true curve, and rolling the plate itself true to template. This will make as perfect a joint as possible. For a large diameter of boiler, I think the joint advocated by Mr. Dean, especially if the edges are scalloped, is an excellent one, but for smaller diameters I prefer the old joint.

4 Two other points must be considered: first, how the calking is done, as in many sheets the initial fracture is caused by bad calking; second, what sort of metal was used, for unless the chemical analysis of the plates as to minimum of injurious metalloids is firmly insisted on, trouble is sure to follow even in the best proportioned joints.

PROF. A. M. GREENE, JR. Mr. Dean is probably aware that in the 1893 report of the Chief of the Bureau of Steam Engineering of the Navy, it is shown that the boilers intended for the New York, the Columbia and the Minneapolis, were all designed on the same plan as that which Mr. Dean recommends. The illustration in the paper is almost exactly similar to those in the report. These boilers were all installed and have given entire satisfaction.

2 Locomotive engineers, however, are using the unequal length butt strap quite extensively. I know of locomotives in which two rows of rivets were placed outside of the outer butt strap, and I do not know of any failure of such joints. If it is a case of getting increased efficiency, and still having the outer butt strap arranged for a calking distance, I do not see why we should depart from the method of unequal straps to use the equal strap arrangement which cannot give such high efficiencies.

WILLIAM A. JONES. I wish to point out the tension which exists in the outer row of rivets and its effect on the drum shell. This should have an important part in determining whether the form of joint which Mr. Dean recommends is really better than if the outer butt strap were cut back one row of rivets on each side, so that the rivets at their calking edges would be close together.

2 We probably all agree that rivets are more reliable in shear than they are in tension; that the more closely and firmly the edge of the outer butt strap is held down, the less calking will be required and

the less possibility there will be of injuring the shell plates by calking the butt strap in the shop, and the more remote will be the probability of subsequent leaks, prompting inexpert men to calk them again later.

3 If we assume that the inner rivets are about 3 in. apart, then the outer rivets shown in the joint which Mr. Dean recommends will be about 6 in. apart, and each rivet will be holding an area of butt strap of from 15 to 20 sq. in., which, at 200-lb. pressure, will require from 3000 to 4000-lb. tension per rivet. In addition, each of these rivets will be required to hold the calking for an edge about 6 in. long, and the calking will have an advantage over the rivet of about 2 to 1, due to the leverage which it has because the rivets are back from the edge. It does not require much thought to see that these rivets would be better able to do this work if they were twice as close together.

4 The joint which Mr. Dean has shown has five rivets in double shear on each side, in a length equal to the pitch of the outer rivets, so that ten times the area of one rivet is the total area in shear in this length. If, on the other hand, the outer butt strap were cut back so that the rivets at its edge would be close together and the outer rivets were in single shear, then the total area in shear would be only one-tenth less, and the proportion of the circular tension transmitted by the rivets in single shear could not be more than 11 per cent of the total in this case.

5 I understand that it is in an effort to improve the action of this 11 per cent of the force involved that this wide outer butt strap is recommended, and that where four rows of rivets are used instead of six, this proportion may rise to 20 per cent. In any case, the slight bending in the shell plate is less, I believe, than the bending tendency which the tension would produce in the rivets, due to pressure on the wide outer butt strap.

6 Let us consider the forces acting upon a rectangular area of plate in a drum shell under pressure. The circular tensions acting tangentially at the edges of this area are equal in intensity, but act at an angle to each other, so that each has a component normal to the chord of the area considered. These normal components exactly balance the pressure acting on that chord. When the area considered embraces a half-circle, the normal components become equal to the circular tension.

7 In the case of the outer butt strap, if all the circular tensions of the drum could be transmitted to the outer butt strap by rivets at its extreme edge, the shear of these rivets alone would hold the outer butt

strap to the drum, and the components of the shears normal to the chord would just balance the steam pressure on that chord, so that no tension in the rivets would be necessary, except for calking. Moving the rivets back from the edge of the butt strap makes the shear act more nearly parallel to the chord, while it does not diminish the chord, so that shear alone will no longer hold the butt strap in place, and tension must be developed in the rivets to make up the difference.

8 Transmitting part of the circular tension through the inside butt strap further increases the tension on the rivets, due to pressure, but the additional tension in this case maintains the curve in the inner butt strap by stitching it to the surface which receives the pressure, and the reaction of the tension at the inner ends of these rivets is thus provided for.

9 In the case of the outer rivets of the joint which Mr. Dean shows, reaction of this tension at the inner ends of the rivets must be absorbed by an abrupt change in direction of the circular tension at those points, tending to produce corners in the drum shell in order to satisfy the triangle of the three forces formed by the tension on the rivet, the tangential tension to the right, and the tangential tension to the left. If we assume a 42-in. drum, 200-lb. steam pressure, 6-in. pitch of outer rivets, each of which takes in tension the pressure of 20 sq. in., we have 4000-lb. tension in each rivet due to steam pressure, the inner ends of the rivets being anchored by an abrupt change in direction of about 9 deg. of 25,200-lb. circular tension.

10 Evidently, this abrupt change of direction of the total circular tension may readily distress the plate more in the form of joint which Mr. Dean recommends than in the usual form of joint with the narrow outer butt strap, even though a very small part of the circular tension is transmitted through a rivet in single shear.

11 Mr. Dean's statement that he believes there has been no case of failure of butt-strap joints, would indicate that there was nothing wrong with the established form using the narrow outer butt strap. Certainly the remedy proposed seems more objectionable than a rivet in single shear.

SHERWOOD F. JETER.[1] It seems that all engineers design joints with reference to their weakest point, that is, provided the joint were to be ruptured in a machine. Of all the explosions that to my knowledge have been due to ruptures, none have occurred in the theo-

[1] The Bigelow Co., New Haven, Conn.

retically weakest part of the joint. Most explosions due to rupture of the sheet have occurred near the joint and were apparently due to flexure of the metal, which had destroyed its life at the particular point of rupture.

2 I believe that there is a great need for an investigation as to what causes the rupture of the plate, and for other than machine tests of different kinds of joints. An account in Power states that there have been four ruptures of butt-strap joints of a nature similar to what was previously alluded to as a "lap cracking" of the joint. From the great number of lap joints in successful use for twenty-five years or more, it may be judged that something besides a mere lapping of the plates causes such defects.

THE AUTHOR. There is very little for me to say in closing, as my views have been fully set forth in the paper. I am interested in the history of this joint as stated by Mr. Bolton. I first knew of it in 1889; it is shown in Thomas W. Traill's book on Boilers, and a table of sizes of parts is there given.

2 Several of the speakers express doubt as to the tightness of the joint on account of the wide spacing of the outer row of rivets. There should be no doubt of this kind, for too many of them are in use. I know of one joint with $1\frac{1}{4}$-in. rivets in 1-in. straps on a pitch of $9\frac{1}{8}$ in., and another with $1\frac{3}{16}$-in. rivets in a $\frac{7}{8}$-in. strap on a pitch of $8\frac{3}{8}$ in.

No. 1329

STRAIN MEASUREMENTS OF SOME STEAM BOILERS UNDER HYDROSTATIC PRESSURE

By James E. Howard,[1] Washington, D. C.

Non-Member

The object of these tests is to ascertain the condition of the metal of the shell and other parts of two horizontal tubular steam boilers which had been in use for a term of service of unusual length; and in addition thereto to acquire information on constructive details by means of measured strains.

2 The boilers were contributed for investigative purposes by the treasurer of the Kendall Manufacturing Company, Providence, R. I., the late Nicholas Sheldon, Esq. They were of early manufacture and from their remarkable history and present condition were of special value for these tests. They were made by the Whittier Machine Company, Boston, Mass., using "Benzon" brand of steel, and were put into service March 1881. They were in continuous service for a period of 27 years, during which time, as Mr. Sheldon wrote, "no repairs were required; in fact, not one cent has been spent upon them."

3 They consisted of five course boilers, two sheets to a course, having the following general dimensions:

Diameter, in.	72
Length over dry sheet, ft.	16
Thickness of shell, in.	$\frac{3}{8}$
Thickness of heads, in.	$\frac{1}{2}$
Number of tubes.	140
Diameter of tubes, in.	3
Length of tubes, ft.	15
Diameter of dome, ft.	$2\frac{1}{2}$

Longitudinal seams, double-riveted lap joints, $\frac{3}{4}$-in. rivets, 2-in. pitch, punched holes, rows $2\frac{1}{8}$ in. apart, rivets staggered.
Girth seam, $\frac{3}{4}$-in. rivets, $2\frac{1}{8}$-in. pitch.
Heads stayed, each, with 14 braces.
Cast-iron manhole frames and safety-valve nozzle.
Supported by lugs, three on a side.
The feedwater came from the Pawtucket River.

[1] Engineer-Physicist, Bureau of Standards, Washington, D. C.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

4 The hydrostatic tests were made at the W. H. Hick's Boiler Works, Providence. Mr. Francis B. Allen, Vice-President of the Hartford Steam Boiler Inspection & Insurance Company, assisted and advised with the writer in conducting them. The boilers will be designated by the numbers 4084 and 4092 under which they were carried on the books of the Hartford company.

5 The tests began with strain measurements upon different parts of the boilers as they were subjected to successive increments of hydrostatic pressures. The results of this portion of the inquiry are now available and herewith presented. Much remains to be done in the other direction of testing, pertaining to the physical properties of the materials.

6 Measurements of the deformations of engineering structures, whether steam boilers, bridges or buildings, may be expected to develop information of a kind not attainable in the tests of the component parts of those structures. A comparatively new field of inquiry is presented in the tests of structures over the tests of the materials thereof. The effects of combined stresses may readily be studied in this manner.

7 No more simple type of boiler could be chosen than the plain horizontal, tubular boiler of these tests, yet it will be seen from the results that complexity of strains and stresses are found in most parts of the shell. In comparatively few places are tangential strains displayed corresponding in magnitude to those which would be expected in a thin cylindrical shell subjected to a given interior pressure.

8 Ascertaining the deformations by the method of measured strains, locally determined, consists of establishing gaged lengths on different parts of the boiler and then measuring them initially and at intervals as the hydrostatic pressures are successively applied and released.

9 Gaged lengths of 10 in. each were used in the examination of these boilers. Their extremities were defined by small drilled and reamed holes. The holes are about 0.05 in. in diameter by, say, 0.10 in. deep, and reamed to a conical shape. The angle of the reamer is 65 deg., and the distance across the hole at the surface of the shell sheet about 0.08 in.

10 Such holes carefully made, in metal surfaces, are capable of centering with considerable precision the contact points of the micrometer strain gage. The strain gage is used as a transfer instrument to compare the gaged lengths on the work with a corresponding length on a standard reference bar.

11 Fig. 1 shows the 10-in. strain gage used on these tests. It consists of two principal parts, an outer tube and an inner stem, which are telescopic, working on ball bearings. Each part carries a conical contact point for centering the instrument on the reference bar and on the work. A screw micrometer measures the length of the instrument when in position.

12 The conical contact points of the strain gage have an angle of 55 deg. This difference of 10 deg. between the reamed hole in the boiler shell and the points of the gage secures contact at a short distance below the surface of the shell. Ordinarily the reference holes are safe against accidental injury, due to their position.

13 As to the degree of precision attained with the strain gage, in the hands of skilled manipulators and under favorable conditions, such as were experienced with these boilers, it is believed the readings are generally reliable to one ten-thousandth of an inch. This strain corresponds to a stress of 300 lb. per sq. in. on a 10-in. gaged length,

FIG. 1 10-IN. STRAIN GAGE

using a modulus of elasticity of 30,000,000 lb. Fig. 2 shows boiler No. 4084. Both boilers were of the same dimensions except at the dry sheets. When on their settings, boiler No. 4084 was on the right, boiler No. 4092 on the left side. This view shows the locations of some of the gaged lengths which were established on this boiler, taken in both tangential and longitudinal directions.

14 A more comprehensive series of lengths was established on boiler No. 4092, and the general discussion of the results of the strain measurements will be given in connection with the test of that boiler.

15 In the test of No. 4084 greater strains were displayed in the vicinity of the dome and the manhole frame than at other parts of the shell. This resulted, as would clearly be expected, in the early failure of the boiler at those places.

16 Actual rupture of the dome was not accomplished, but leakage along its single-riveted longitudinal seam became so great at 266 lb. pressure that it was necessary to remove the dome and patch the shell in order to reach higher pressures with the pump available.

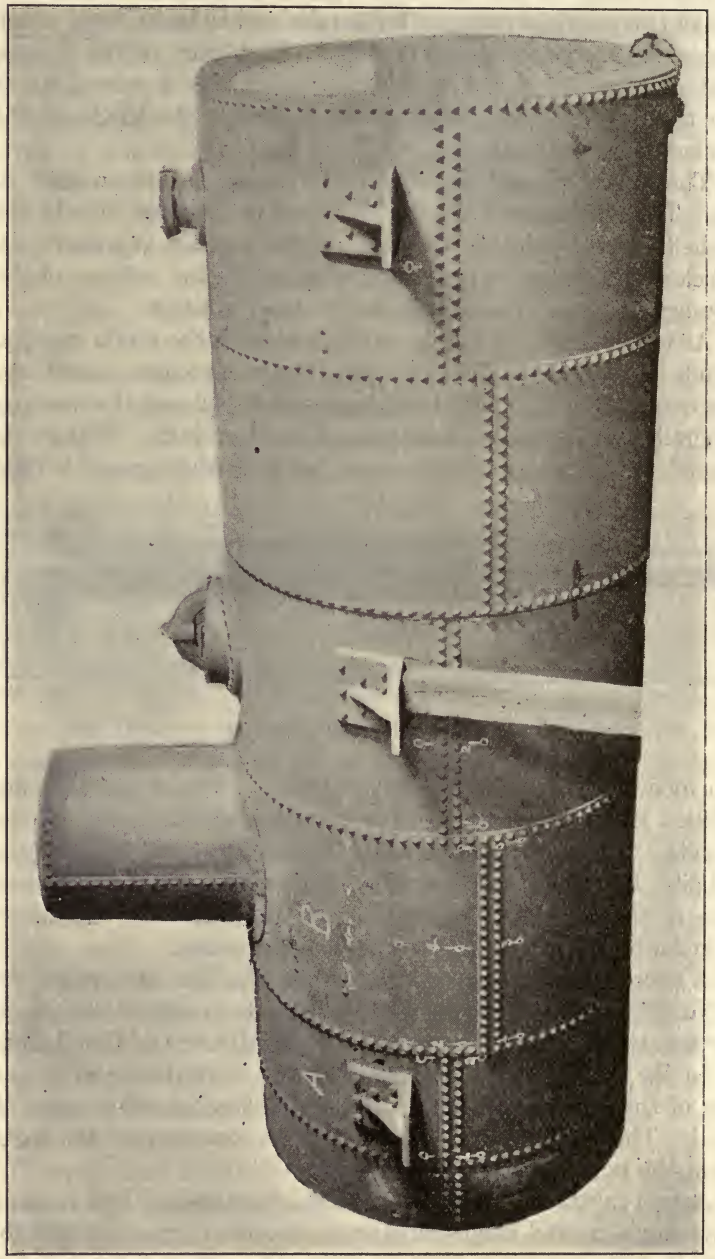


FIG. 2 STEAM BOILER NO. 4084 BEFORE TESTING

17 At 270 lb. pressure the cast-iron manhole frame fractured across the middle of its length. Another patch was then put on the shell covering the manhole.

18 The test was again resumed when at 295 lb. pressure the rupture of three braces of the front head occurred. The test was then discontinued and the boiler dismantled.

FIG. 3 INTERIOR OF DOME SHOWING LINES ALONG WHICH SCALE WAS DISTURBED AFTER PRESSURE OF 266 LB. ON SHELL

19 The strain measurements made in the test of No. 4084 were of the same general order as those subsequently made in the test of the second boiler and the results were, for the most part, quite similar.

20 A feature, however, in the test of the first was absent or obscure in that of the second. There was a progressive difference in the extensibility taken across the longitudinal seams of the several courses in passing from the front to the rear end of the boiler.

TABLE 1 TANGENTIAL EXTENSIONS OF THE SEAMS, BOILER NO. 4084

Pressures	Course			
	B	*C*	*D*	*E*
210	0.0166	0.0121	0.0099	0.0084
240	0.0241	0.0171	0.0138	0.0121
270	0.0341	0.0241	0.0212	0.0187

21 The tangential extensions of the seams, at the middle of their lengths, were as shown in Table 1.

22 While these seams were not directly exposed to the heated gases over the grate, nevertheless it seems probable that a wider range of thermal conditions prevailed in the vicinity of the seams at the front end over those at the rear end of the boiler. If such was the case it would aid in explaining the greater slip of the forward seams.

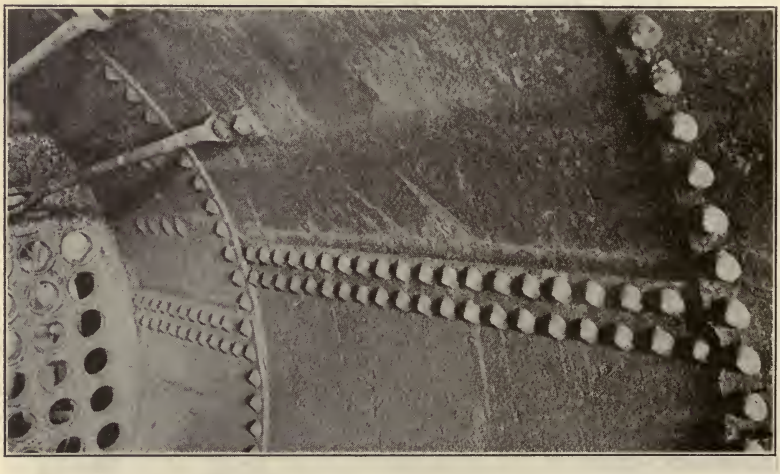


FIG. 4 INTERIOR VIEW OF BOILER NO. 4084 AFTER 295 LB. PRESSURE SHOWING SCALE DISTURBED IN VICINITY OF THE LONGITUDINAL SEAMS

23 Hydrostatic pressures on the exterior surfaces of the tubes necessarily extend them in length. The amount of the extension appears to depend upon their position with reference to their proximity to the shell. Tubes adjacent to the shell extended less than those at the middle of the rows, a restraining influence from the shell appearing to affect the outer ones.

24 The results in Table 2 were obtained by measuring the tubes over their full length.

25 Practically no leakage occurred about the tubes throughout the test of this boiler. A slight leakage took place at two tubes at 120 lb. pressure, but soon ceased and was not renewed during the remainder of the test. The girth seams remained tight up to 210 lb. pressure, and then showed only small leaks which were not materially increased under the higher pressures.

26 Leakage at the longitudinal seams began at 120 lb. pressure and increased as higher pressures were applied. The leakage became general at these seams with 180 lb. pressure on the boiler, but at

this time the slip of the joints had become a pronounced feature of the case, which necessarily disturbed the calking.

TABLE 2 EXTENSION OF TUBES, BOILER NO. 4084

Pressures	Third Row		Seventh Row	
	Next Shell	Middle of Row	Next Shell	Middle of Row
210	0.0110	0.0162	0.0077	0.0167
240	0.0128	0.0184	0.0086	0.0189
270	0.0142	0.0210	0.0099	0.0210

27 Upon removal of the dome, evidence of overstraining was found at its base next the flanged portion. The scale had been disturbed on the inside on the line and in the vicinity of the upper element of the boiler, as shown in Fig. 3. Near the flange the scale was disturbed in oblique, shearing directions, which changed to longitudinal and then tangential directions a little farther up the dome.

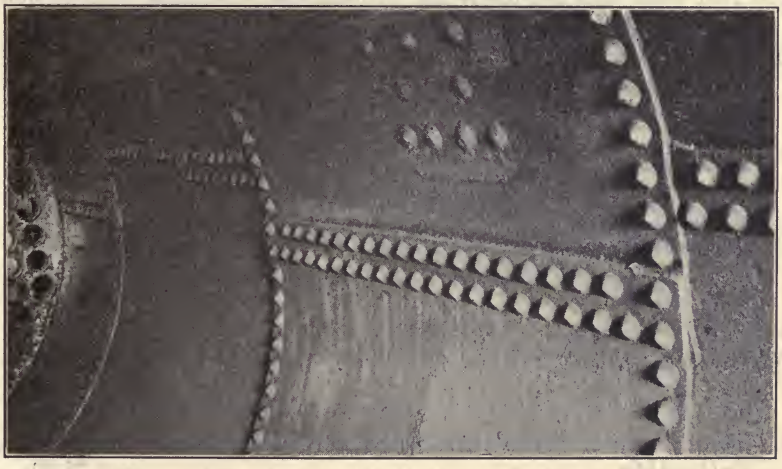


Fig. 5 Interior View of Boiler No. 4084 after 295 Lb. Pressure Showing Scale Disturbed in Vicinity of the Longitudinal Seam and under Supporting Lug

28 Figs. 4 and 5 are interior views of the boiler illustrating the manner in which the scale was disturbed during the test in the vicinity of the longitudinal seams and under one of the lugs.

29 Struts were used under the middle lugs and supported a part of the weight of the boiler during the test. They probably intensi-

fied the stresses in the shell in that vicinity. The interior surface of the shell and also the heads were found in good condition. Fig. 6 shows the appearance of the inside of the rear head.

30 A series of six photographs, Figs. 7 to 12 inclusive, shows the appearance of the exterior surfaces of the tubes. The system of lettering and numbering the horizontal and the vertical rows is indicated in Fig. 6. The layout of the feed pipes appears on Fig. 13.

31 Tubes in the horizontal row marked *D*, the fourth in the boiler from the top, Fig. 7, had a deposit on the rear third of their length, and also a slight deposit on the front ends. On some of the lower rows the deposit was thicker, as Figs. 8–12 indicate. In general the deposit was greatest in the lower rows of tubes and on those farthest from the shell, being confined chiefly to the rear quarter or half of their lengths. The lower rows of tubes, at the front end of the boiler, had a deposit on them. The surfaces of the upper row and the side rows were clean without deposit.

32 Material collected from the bottom of the boiler at the front and rear ends had the following approximate chemical composition:

Deposit from front end of boiler:	Per Cent
Loss on ignition	26.00
SO_3	4.10
Silica	21.60
$Fe_2O_3+Al_2O_3$	25.40
Lime	5.30
Magnesia	13.10
Copper oxide (about)	0.25
CO_2	slight amount
Chlorides	trace

Deposit from rear end of boiler:	Per Cent
Loss on ignition	23.85
SO_3	trace
Silica	27.60
$Fe_2O_3+Al_2O_3$	24.80
Lime	9.60
Magnesia	12.70
Copper oxide (about)	0.25
Chlorides	trace
Carbonates	small amount

33 Prior to testing boiler No. 4092, it was stripped of its dome and manhole frame and the shell patched at those places. The heads were strengthened by means of six $1\frac{1}{4}$-in. braces extending from head

to head. The cast-iron safety-valve nozzle was allowed to remain in place, but was eventually replaced by a soft patch, after 300 lb. pressure had been applied and released. The distortion of the shell under the flange of the nozzle caused leaks impracticable to calk.

34 Fig. 14 shows the boiler when about ready for the hydrostatic test. It was supported on two wooden shoes sawed to fit the curvature of the shell, in lieu of the blocking shown in the illustration.

35 The gaged lengths which were established on the right side

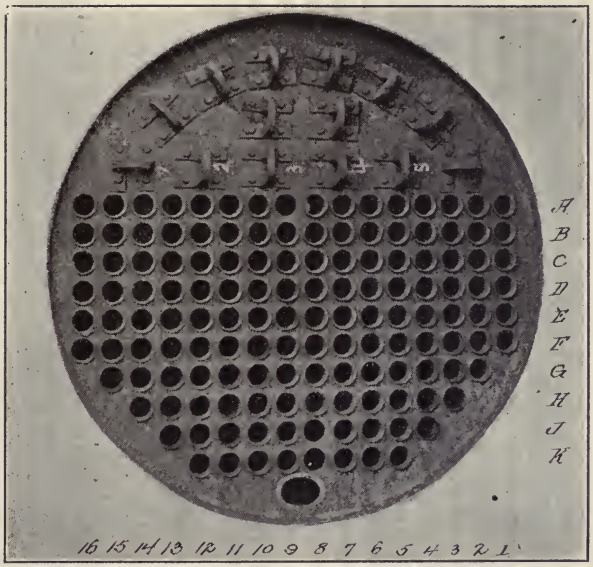


FIG. 6 INSIDE OF REAR HEAD, BOILER NO. 4084, AFTER 295 LB. PRESSURE. LETTERS AND FIGURES REFER TO MARKS SHOWING LOCATION OF TUBES IN SUBSEQUENT PHOTOGRAPHS

of the boiler are shown in this figure and practically stand for those on the left side, while additional ones were laid off on the top. Figs. 15, 16 and 17 show, however, the different gaged length on each side and the top. Additional gaged lengths were laid off and measured which are not shown on these diagrams; their results confirm those which will be referred to. There were 165 gaged lengths used in the principal series of observations, on which some 3300 readings were taken.

36 The general results of the strain measurements have been plotted on a series of ten diagrams, Figs. 18 to 27, inclusive. For the

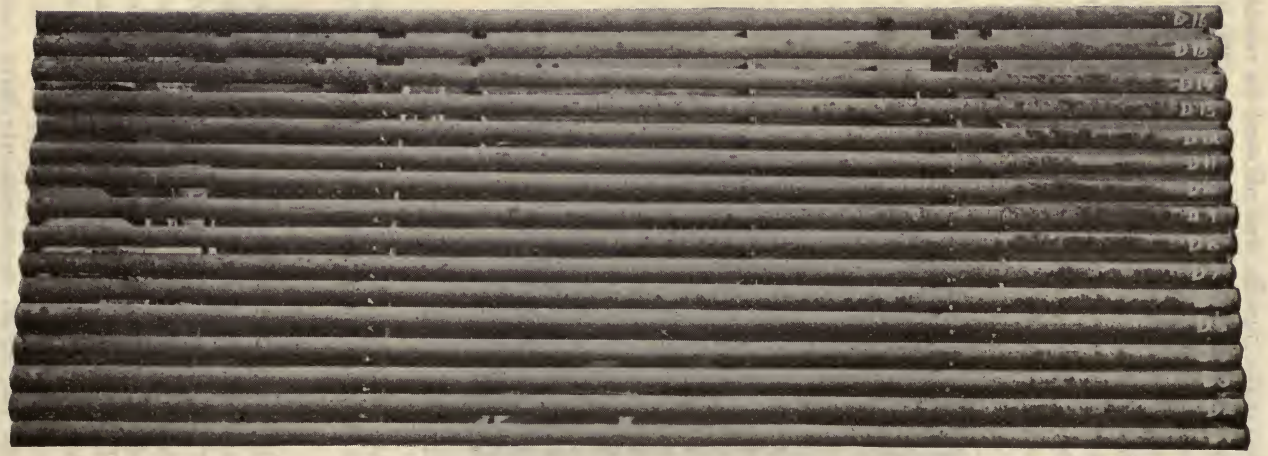


FIG. 7 APPEARANCE OF TUBES, BOILER NO. 4084, AFTER DISMANTLING. HORIZONTAL ROW *D*, FOURTH FROM TOP

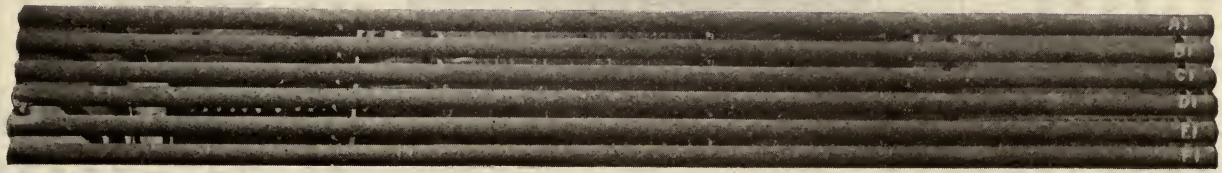


FIG. 8 APPEARANCE OF TUBES, BOILER NO. 4084, AFTER DISMANTLING. VERTICAL ROW, FIRST ON RIGHT SIDE OF BOILER

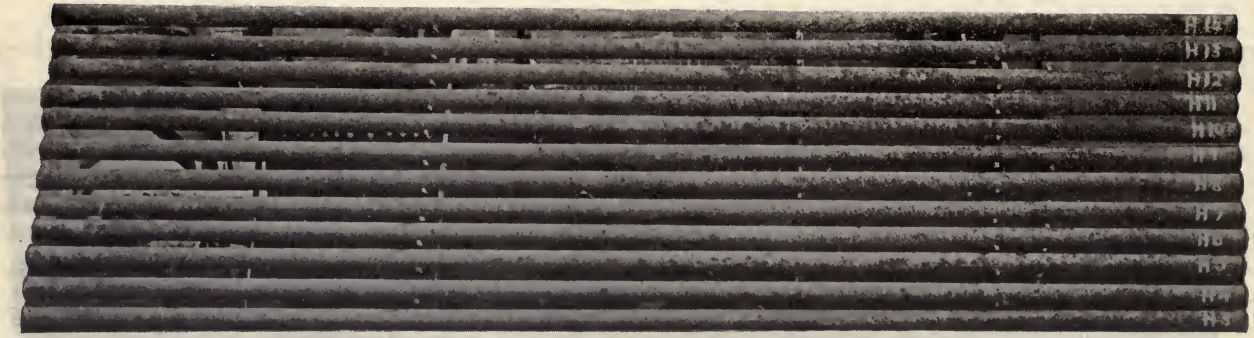


FIG. 9 APPEARANCE OF TUBES, BOILER NO. 4084, AFTER DISMANTLING. HORIZONTAL ROW *H*, EIGHTH FROM TOP

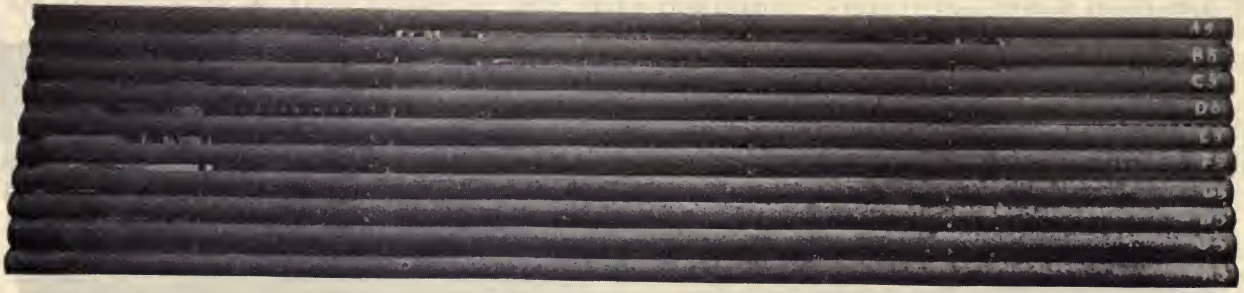


FIG. 10 APPEARANCE OF TUBES, BOILER NO. 4084 AFTER DISMANTLING. VERTICAL ROW, FIFTH FROM RIGHT SIDE OF BOILER

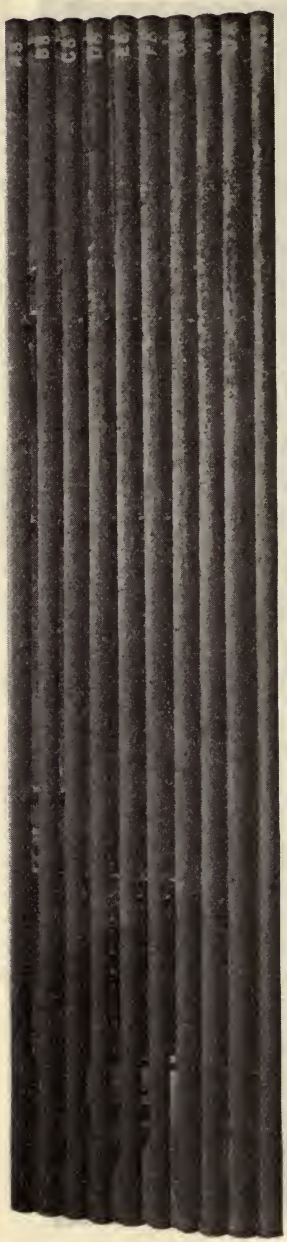


FIG. 11 APPEARANCE OF TUBES, BOILER NO. 4084, AFTER DISMANTLING. VERTICAL ROW, EIGHTH FROM RIGHT SIDE OF BOILER

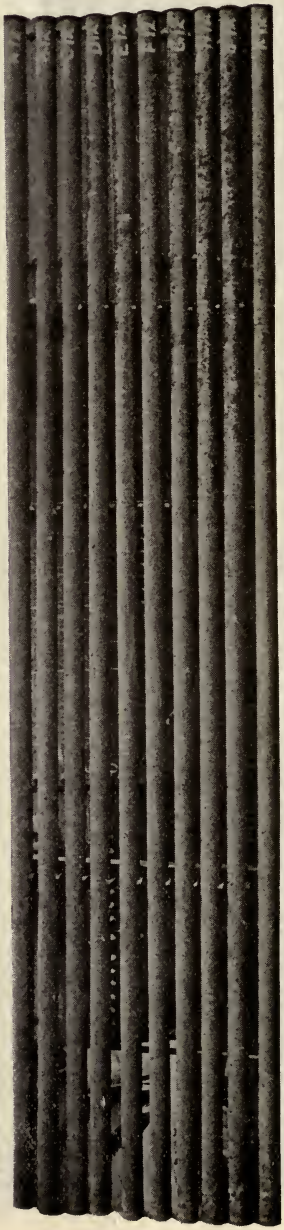


FIG. 12 APPEARANCE OF TUBES, BOILER NO. 4048, AFTER DISMANTLING. VERTICAL ROW, TWELFTH FROM RIGHT SIDE OF BOILER

purpose of furnishing a convenient basis of comparison, in judging the behavior of the boiler at different parts and under different pressures, heavy lines have been drawn on each diagram which indicate strains corresponding to those which would be displayed by the

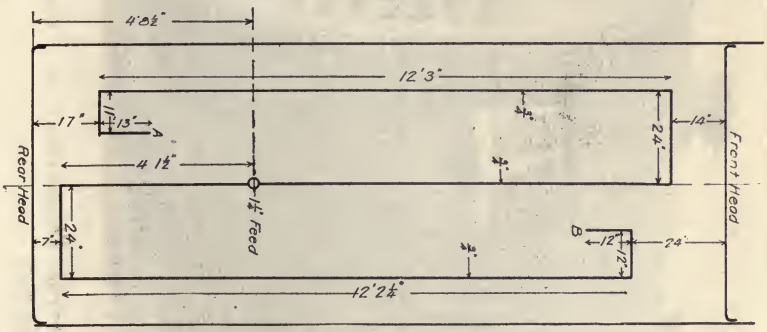


FIG. 13 LAYOUT OF FEED-PIPES
Pipe *A* Discharged directly over Tubes of Row No. 5 at Rear End
Pipe *B* Discharged directly over Tubes of Row No. 12 at Front End

TABLE 3 COMPUTED STRESSES ON THE SHELL SHEETS, BOILER NO. 4092

Boiler Pressure, Lb. per Sq. In.	Stress on ½-In. Shell, Lb. per Sq. In.	Strain on Gaged Length of 10 In.
30	2880	0.0010
60	5760	0.0019
90	8640	0.0029
120	11520	0.0038
150	14400	0.0048
180	17280	0.0058
210	20160	0.0067
240	23040	0.0077
270	25920	0.0086
300	28800	0.0096

sheets under direct tensile stresses, using a modulus of elasticity of 30,000,000 lb. per sq. in. Plotted curves, which are steeper than the modulus of elasticity reference line, indicate places on the boiler having greater rigidity than normal to the plain sheets; while flatter curves indicate greater extensibility than pertains to the plain metal.

37 Tangential rigidity above the normal was displayed in the vicinity of the girth seams, while the gaged lengths taken across the longitudinal seams at the middle of the length of the courses showed a much lower degree of rigidity than common to the plain sheet. Zones of greater extension than normal were also found in the vicinity of the manhole and dome.

FIG. 14 STEAM BOILER, NO. 4092, DOME AND MANHOLE FRAME REMOVED PREPARATORY TO TESTING. HEADS STRENGTHENED WITH 6 THROUGH BRACES

38 Flattening of the curves representing the solid sheets necessarily accompanied those pressures which caused a tensile stress on the shell in excess of its elastic limit.

39 Table 3 shows the computed stresses on the shell sheets, considering only tangential stresses as acting, and the strains which should

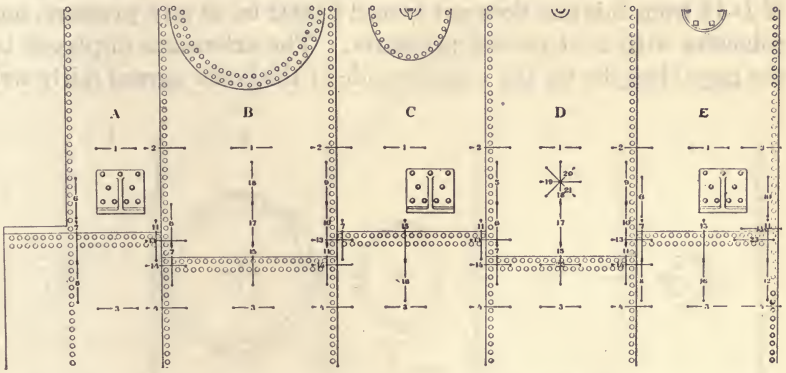


FIG. 15 DIAGRAM SHOWING LOCATION OF GAGED LENGTHS, 10 IN. EACH, LAID OFF ON RIGHT SIDE ON BOILER NO. 4092

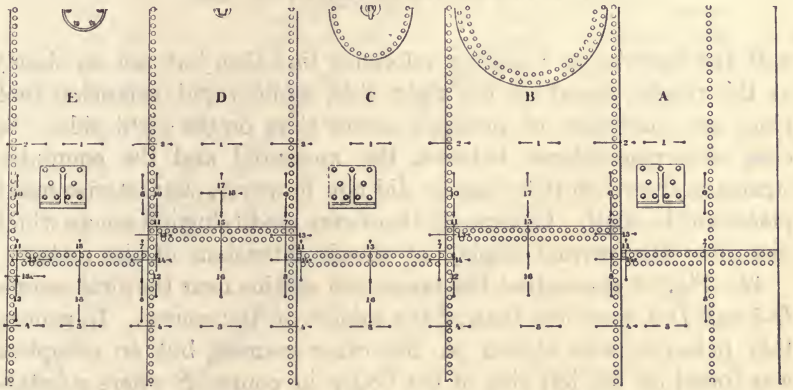


FIG. 16 DIAGRAM SHOWING LOCATION OF GAGED LENGTHS, 10 IN. EACH, LAID OFF ON LEFT SIDE OF BOILER NO. 4092

be developed on a gaged length of 10 in., using a modulus of elasticity of 30,000,000 lb., the interior diameter of the boiler being 72 in.

40 Referring now to the plotted results, Fig. 18 shows the tangential extensions of sheets *C* and *D* on the right and left sides of the

boiler respectively, taken at the middle of the lengths of the courses. Gaged length *D*-18, on the right side of the boiler, was located above the longitudinal seam, while *C*-16, on the right side, was located below the longitudinal seam.

41 The tangential extensions of each of these gaged lengths closely follow the modulus of elasticity comparison curve. The departure of *D*-18 from this line does not exceed 0.0002 in. at any pressure, and coincides with it at several pressures. The extensions displayed by the gaged lengths on the opposite side of the boiler agreed fairly well

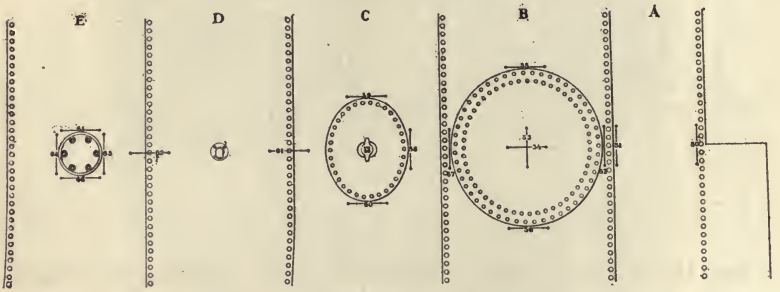


FIG. 17 DIAGRAM SHOWING LOCATION OF GAGED LENGTHS, 10 IN. EACH, LAID OFF ON TOP OF BOILER NO. 4092

with the modulus of elasticity reference line also, but not so closely as the results found on the right side, while rapid extension took place one increment of pressure earlier than on the right side. So close a correspondence between the measured and the computed strains as shown on this diagram did not, however, characterize many places on the shell. Commonly there were modifying influences which disturbed the normal display of elastic extensions of the metal.

42 Fig. 19 shows that the tangential strains near the girth seams, *D*-5 and *D*-9, were less than at the middle of the course. In general this behavior was shown in the other courses, but an exception was found on the left side of the boiler in course *B* where substantially the same rigidity was displayed at the middle as at the edges of the course.

43 In the third group of curves on this diagram, however, the extension of *D*-16 taken at the middle of the course is seen to be greater than at *D*-8 and *D*-12, curves representing the edges.

44 There is a marked difference in the tangential extension of the two edges of the end courses of the boiler, due to the influence of the

heads in supporting the shells. Fig. 20 shows the greater rigidity of gaged lengths E-12 which are taken nearly over the rear head, than at the other places, which were measured on this course.

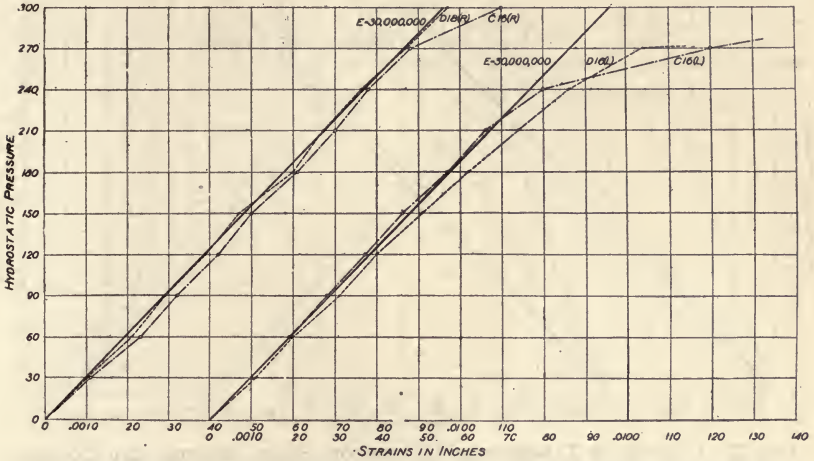


FIG. 18 CURVES OF TANGENTIAL EXTENSION, SOLID SHEETS, AT MIDDLE OF LENGTH OF COURSES C AND D, RIGHT AND LEFT SIDES OF BOILER

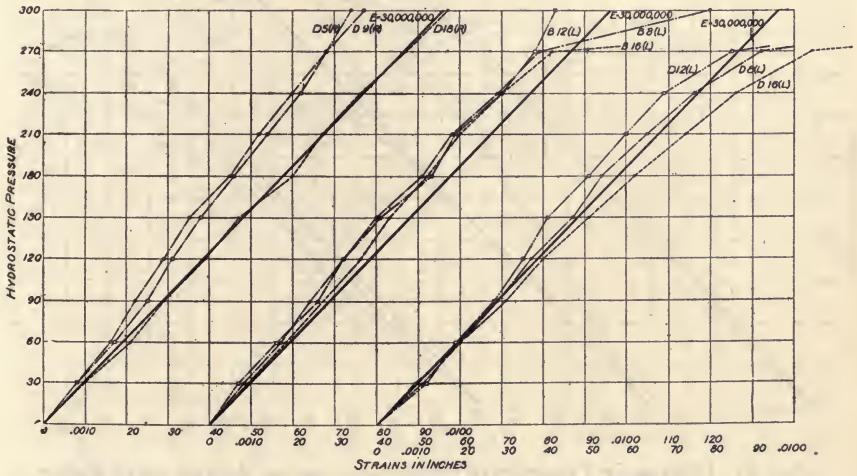


FIG. 19 CURVES OF TANGENTIAL EXTENSION, SOLID SHEETS, NEAR GIRTH SEAMS AND AT MIDDLE OF LENGTHS OF COURSES

45 In regard to the top of the boiler there were many disturbing factors present, as indicated by Fig. 21. The first course is a short

one with the front head to stiffen one edge. Then came the dome in the original construction on course *B*, which was patched, and the

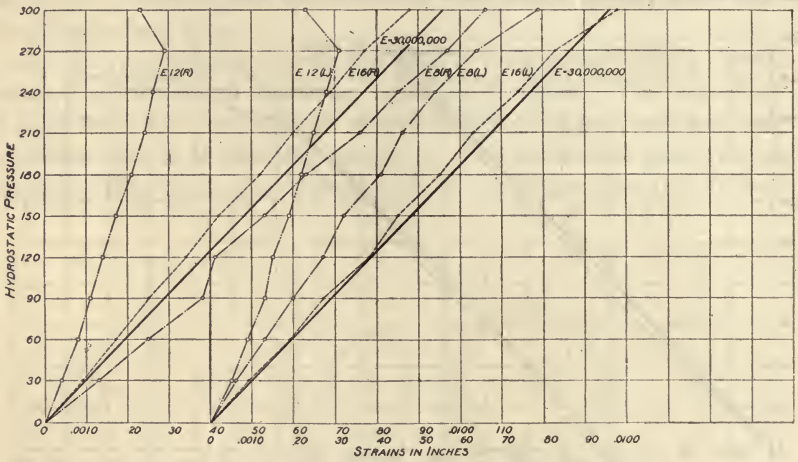


FIG. 20 CURVES OF TANGENTIAL EXTENSION, SOLID SHEETS, END COURSE *E*, NEAR REAR HEAD, GIRTH SEAM, AND MIDDLE OF LENGTH OF COURSE

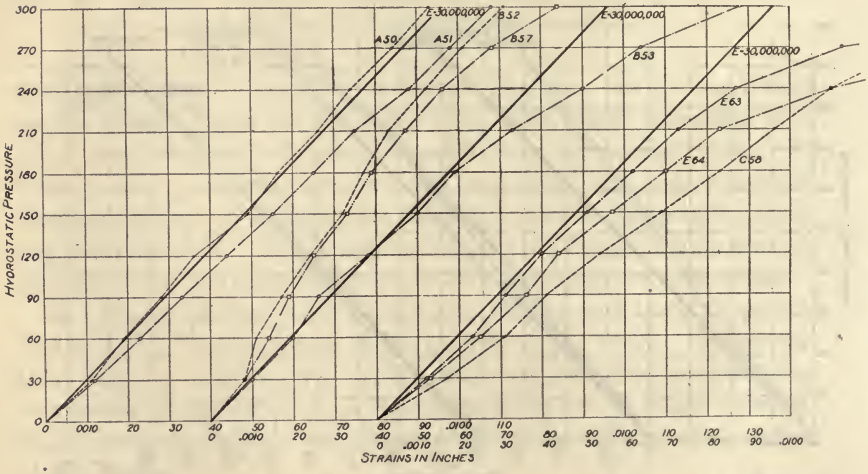


FIG. 21 CURVES OF TANGENTIAL EXTENSION, TOP OF BOILER, NEAR FRONT HEAD, GIRTH SEAMS, DOME AND MANHOLE PATCHES, AND SAFETY-VALVE NOZZLE

patch double-riveted, using the rivet holes that were made for securing the flange of the dome to the shell. Course *C* had the manhole

patch, a single riveted one, using the holes made for securing the manhole frame to the shell. In course *D* was found the feedwater

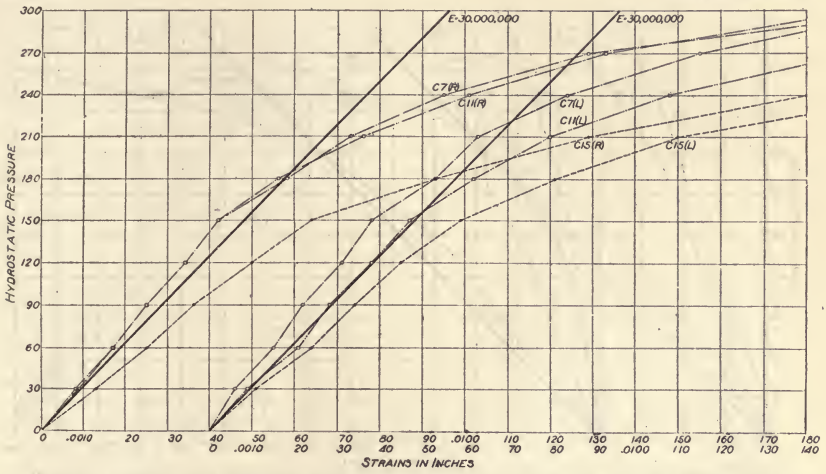


FIG. 22 CURVES OF TANGENTIAL EXTENSION, ACROSS LONGITUDINAL SEAMS, AT MIDDLE OF LENGTH OF COURSE *C*, AND AT EDGES, RIGHT AND LEFT SIDES OF BOILER

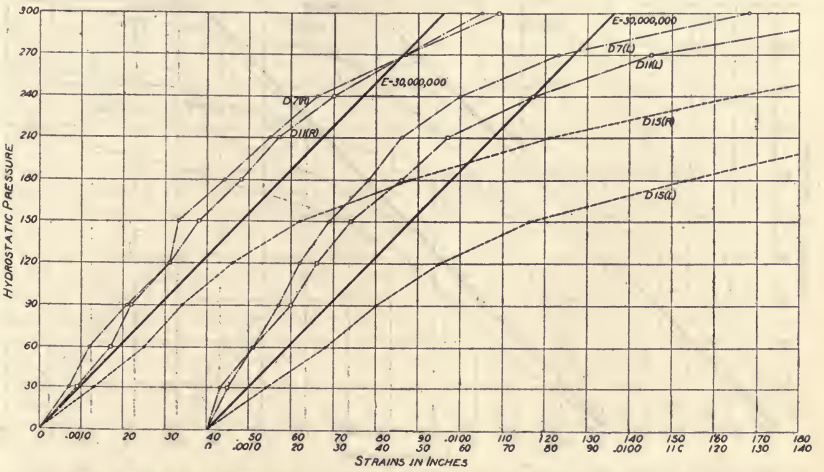


FIG. 23 CURVES OF TANGENTIAL EXTENSION, ACROSS LONGITUDINAL SEAMS, AT MIDDLE OF LENGTH OF COURSE *D*, AND AT EDGES, RIGHT AND LEFT SIDES OF BOILER

connection which probably did not have much influence on the behavior of this course while under pressure. Course *E* had the cast-iron

safety-valve nozzle riveted to it, which did seem to have an influence on the tangential extension of the steel, permitting greater extension than normal.

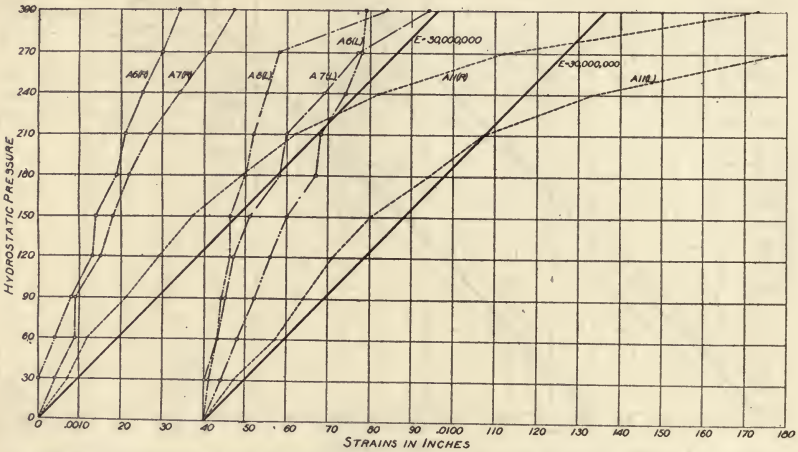


FIG. 24 CURVES OF TANGENTIAL EXTENSION, ACROSS LONGITUDINAL SEAMS AND SOLID SHEETS, END COURSE *A*; EXTENSIONS NEAR FRONT AND GIRTH SEAM

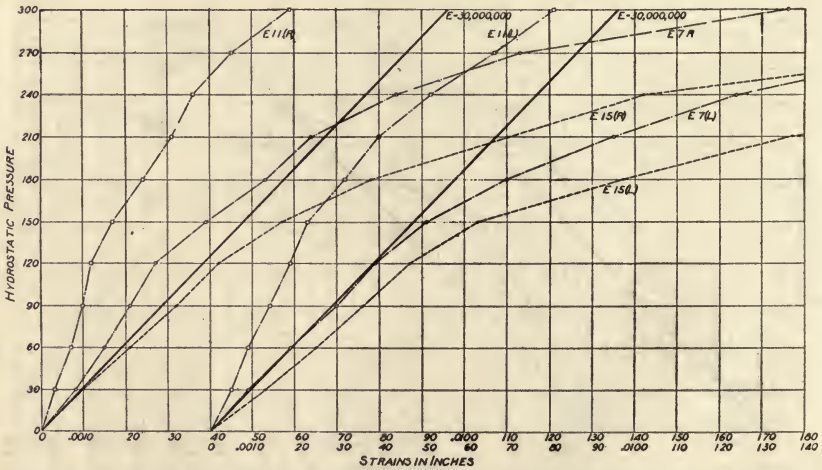


FIG. 25 CURVES OF TANGENTIAL EXTENSION, ACROSS LONGITUDINAL SEAMS, END COURSE *E*, EXTENSIONS NEAR REAR HEAD AND GIRTH SEAM AND MIDDLE OF LENGTH OF COURSE

46 The extension of course *A* at the edge over the front head was less than at the opposite edge. Gaged lengths *B*-52 and *B*-57 showed

the influence of the overlapping metal of the patch, as well as that of the girth seams. The extension at these two places was less than normal.

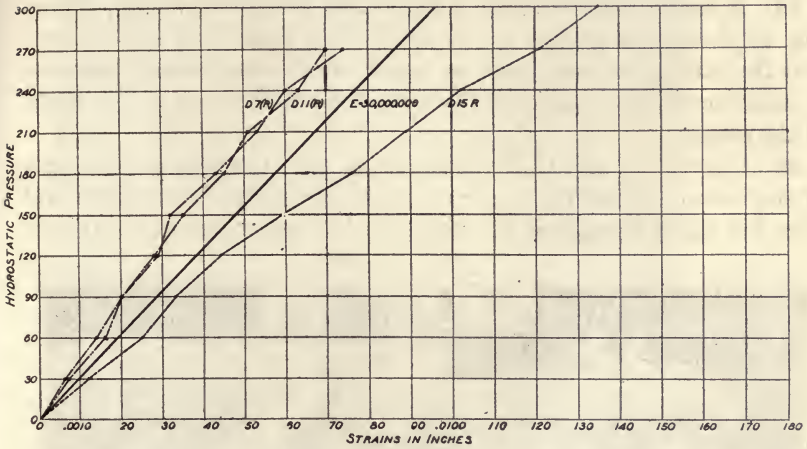


FIG. 26 CURVE OF TANGENTIAL RESILIENCE, ACROSS LONGITUDINAL SEAM, COURSE *D*, RIGHT SIDE OF BOILER, AT MIDDLE OF COURSE AND NEAR GIRTH SEAMS

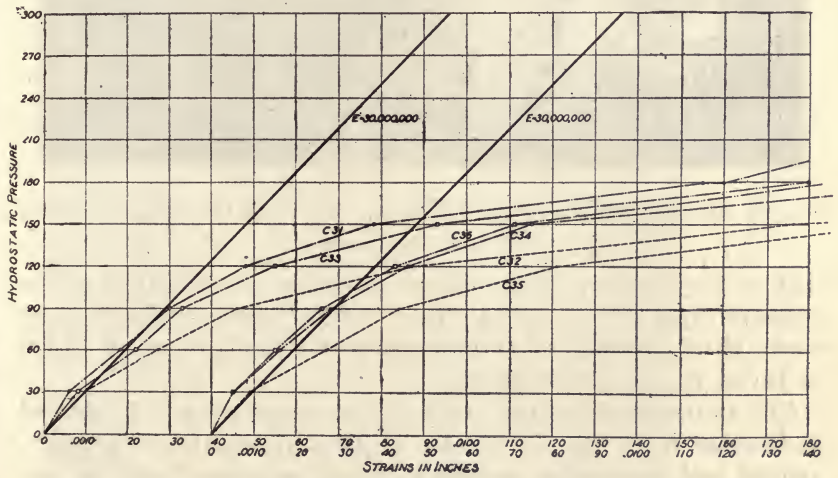


FIG. 27 CURVES OF TANGENTIAL EXTENSION, HAND-RIVETED SECTION, COURSE *C*, TOP OF BOILER

47 At the side of the manhole patch, *C*-58, there was found diminished rigidity in the shell. The weakness of this single-riveted patch was apparent in the measurements from the earliest pressures which

were applied to the boiler. Conditions about the safety-valve nozzle did not seem fully to compensate for this opening in the shell, as shown by the extensions on gaged lengths *E*-63 and *E*-64.

48 Referring next to the behavior of the shell at the seams, Fig. 22 shows two groups of curves of three lines each which represent the tangential extensions on gaged lengths established on course *C* taken across the longitudinal seam at the middle and at the edges of the course.

49 Curves *C*-7 and *C*-11, representing the extension at the edges of the course on the right side of the boiler, coincide in places and show but slight divergence where they depart most from the same line;

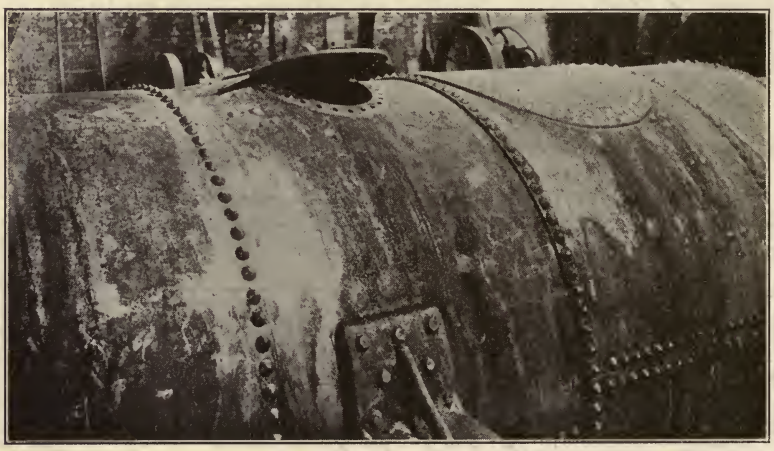


FIG. 28 MANHOLE PATCH AFTER RUPTURE. MAXIMUM PRESSURE ON SHELL 335 LB. PER SQ. IN.

that is, they indicate that uniform behavior was displayed at the opposite edges of this course. Each deflect rapidly under pressure above 150 lb. per sq. in., corresponding to a tensile stress of 14,400 lb. per sq. in. on the solid sheet.

50 At the middle of the length of the course, curve *C*-15 showed an increase in the rate of extension at the above-mentioned pressure, and for each succeeding pressure a greater extension than that witnessed at the edges. Necessarily, variations in the tangential extensions at different parts of the length of a seam would cause variations in the stresses of the solid metal of the shell in those localities.

51 In the case of a three-course boiler with one sheet to a course, as found in current construction, it would seem that a double-riveted

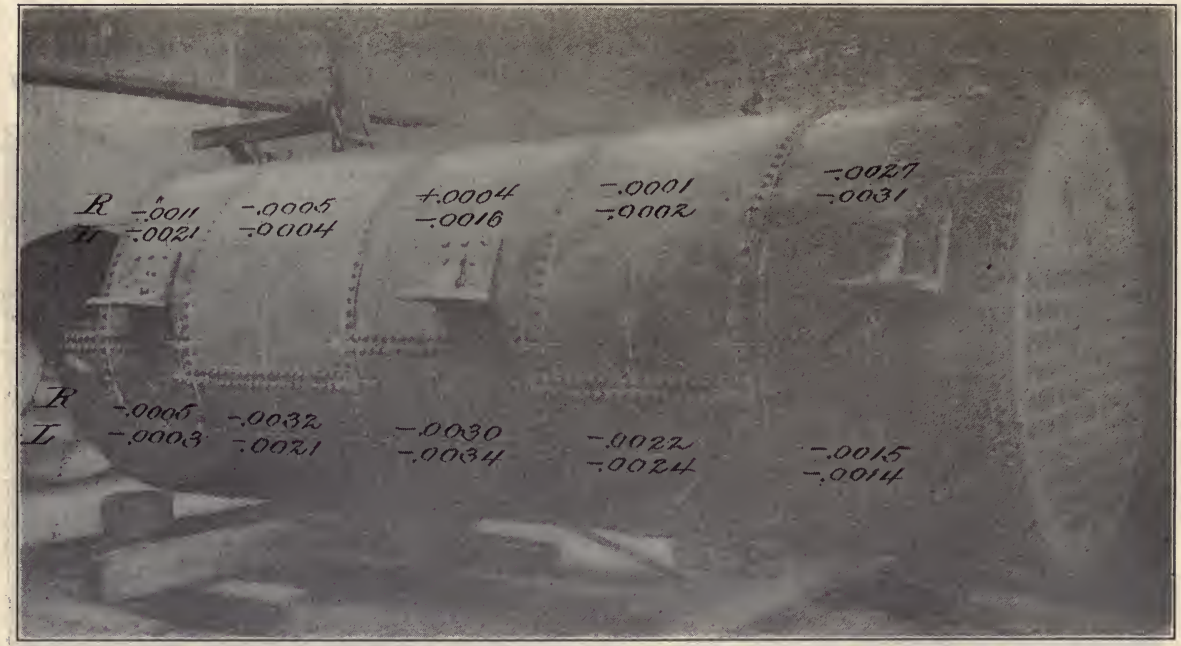


FIG. 29 LONGITUDINAL STRAINS ON SOLID SHEETS AT MIDDLE OF LENGTH OF COURSES, AT 270 LB. PRESSURE PER SQ. IN., RIGHT AND LEFT SIDES OF THE BOILER. MINUS SIGNS INDICATE LONGITUDINAL CONTRACTION; PLUS SIGNS LONGITUDINAL EXTENSION

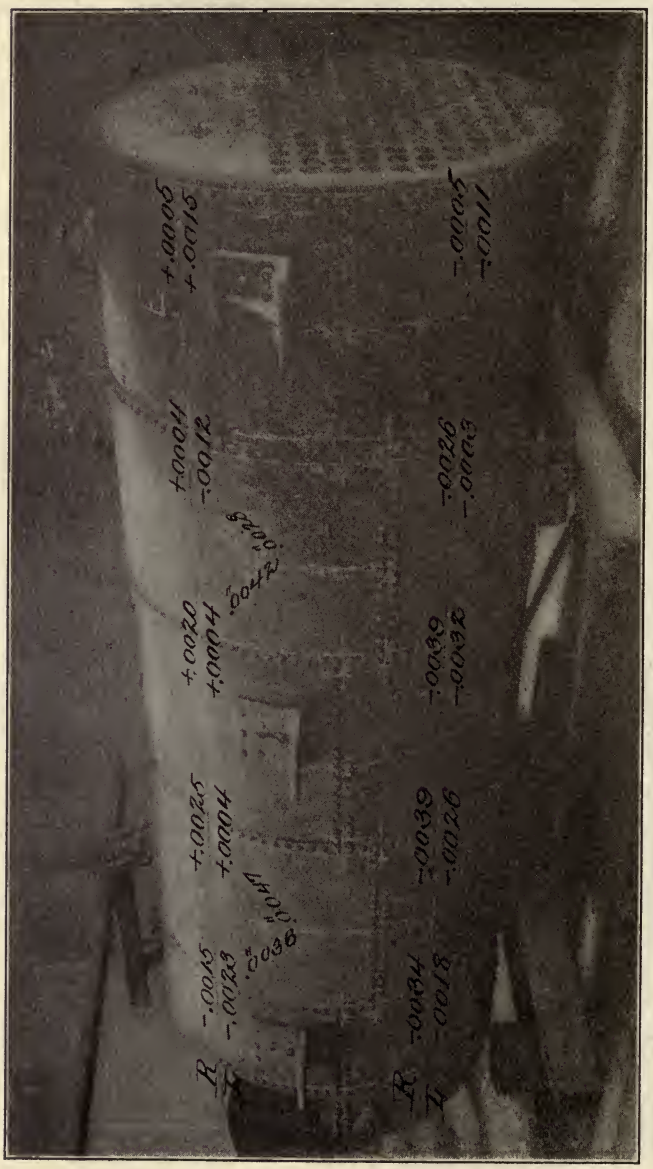


FIG. 30 LONGITUDINAL STRAINS, ACROSS GIRTH SEAMS, AT 270 LB. PRESSURE PER SQ. IN., RIGHT AND LEFT SIDES OF THE BOILER. ALSO STRAINS ON DIAGONAL GAGED LENGTHS OF COURSES *B* AND *D*. MINUS SIGNS INDICATE LONGITUDINAL CONTRACTION: PLUS SIGNS LONGITUDINAL EXTENSION

lap joint might occasion an excessive stress in the solid sheet abreast the end of the seam, under certain pressures.

52 In the present test the longitudinal seams, being only three rivet pitches apart, furnish a line from front to rear of the boiler across which the extensions are greater than those which are displayed by the solid sheets.

53 Fig. 23 shows results corresponding to those of Fig. 22 but pertaining to *D*, the next course of the boiler. The results are about the same on each, the maximum tangential extensions being displayed at the middle of the length of the seams.

54 Fig. 24 shows the extension of end course *A*, across the longitudinal seams on either side of the boiler and also the extension of

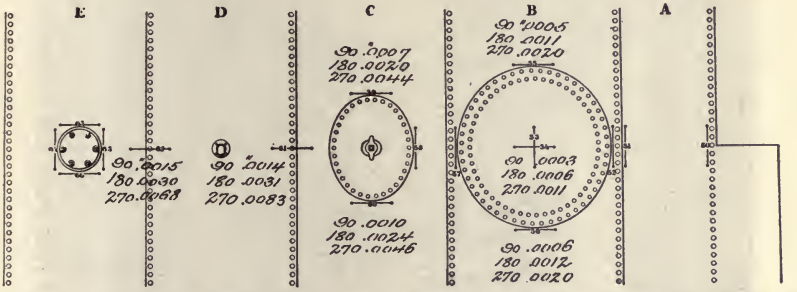


FIG. 31 LONGITUDINAL STRAINS, ON TOP OF BOILER IN VICINITY OF DOME AND MANHOLE PATCHES, ON DOME PATCH, AND ACROSS GIRTH SEAMS, AT 90, 180 AND 270 LB. PRESSURE

the solid sheet near the girth rivets of the front head. The divergent curves of this diagram indicate how differently in degree the metal is strained at the several gaged lengths of this narrow course. There seems, however, no lack of consistency in the behavior of the metal. The strains were relatively such as would be expected under the conditions present in this part of the boiler.

55 The strains in course *E*, at the rear end of the boiler are shown in Fig. 25, where the behavior of the shell was found to be similar to that at the front end. Notwithstanding the fact that the results appear consistent and the relations between the different parts of the boiler harmonious, attention is attracted by the variableness of the strains as they are found developed, according to the position of the measured lengths. The degree of variability witnessed in this type of boiler, which is certainly one of plain form, is such as to excite speculative interest in more complicated types.

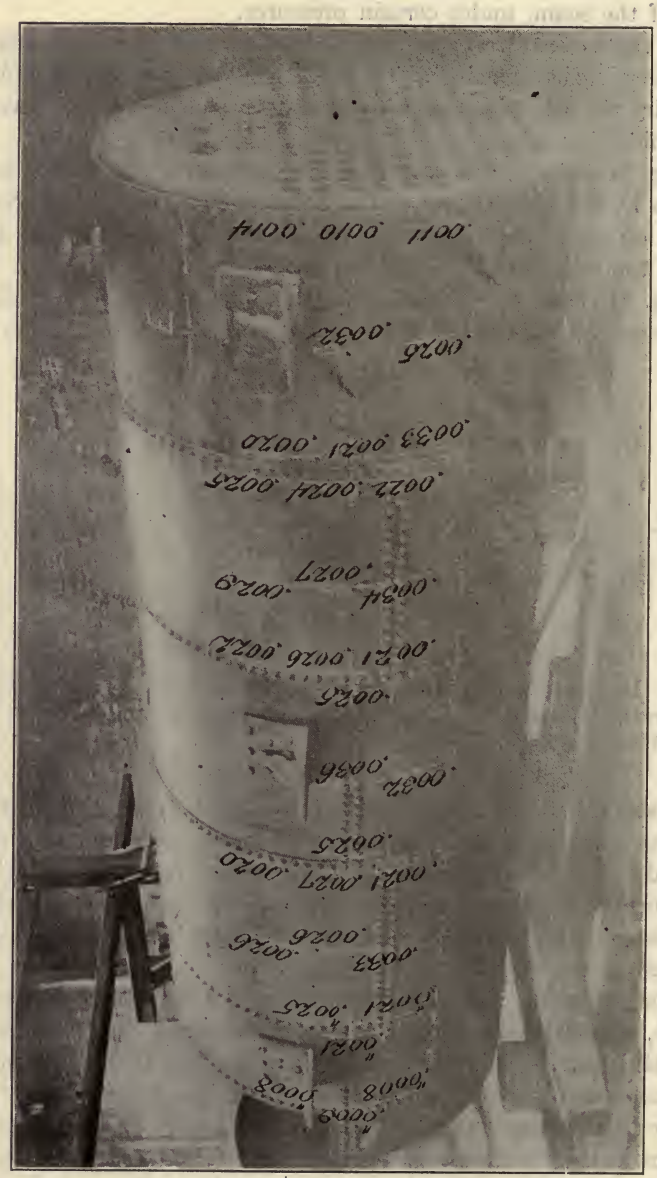


FIG. 32 TANGENTIAL EXTENSION AT 90 LB. PRESSURE. RIGHT SIDE OF BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0029 IN.

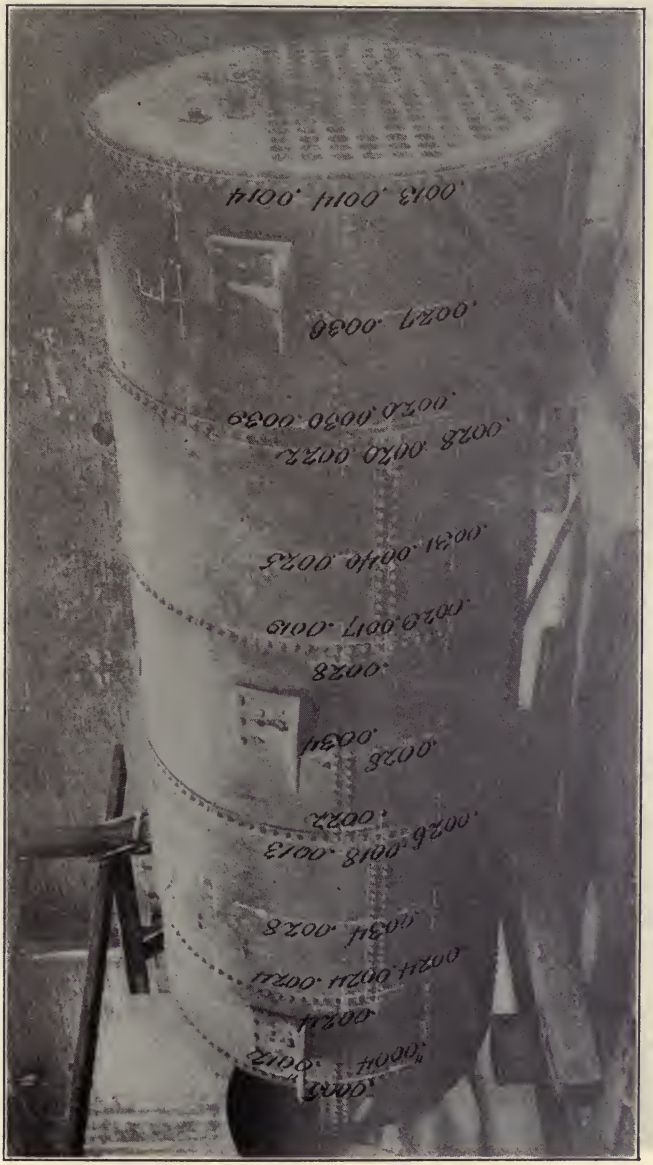


FIG. 33 TANGENTIAL EXTENSION AT 90 LB. PRESSURE. THE FIGURES GIVEN REPRESENT STRAINS DEVELOPED ON THE LEFT SIDE OF THE BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0029 IN.

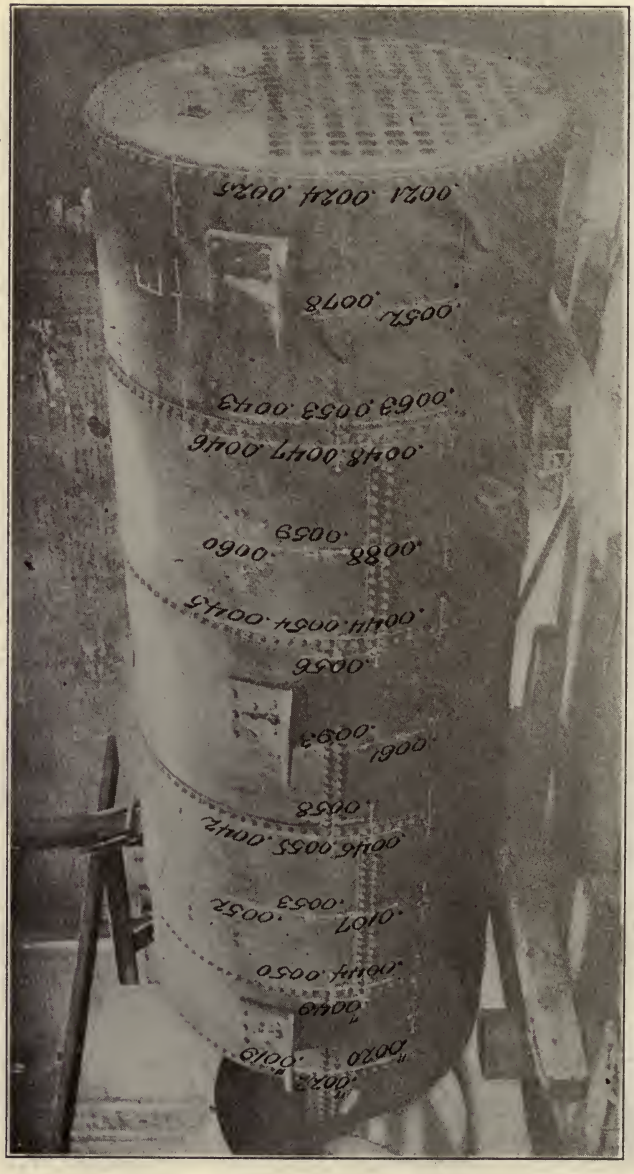


FIG. 34 TANGENTIAL EXTENSION AT 180 LB. PRESSURE. RIGHT SIDE OF BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0058 IN.

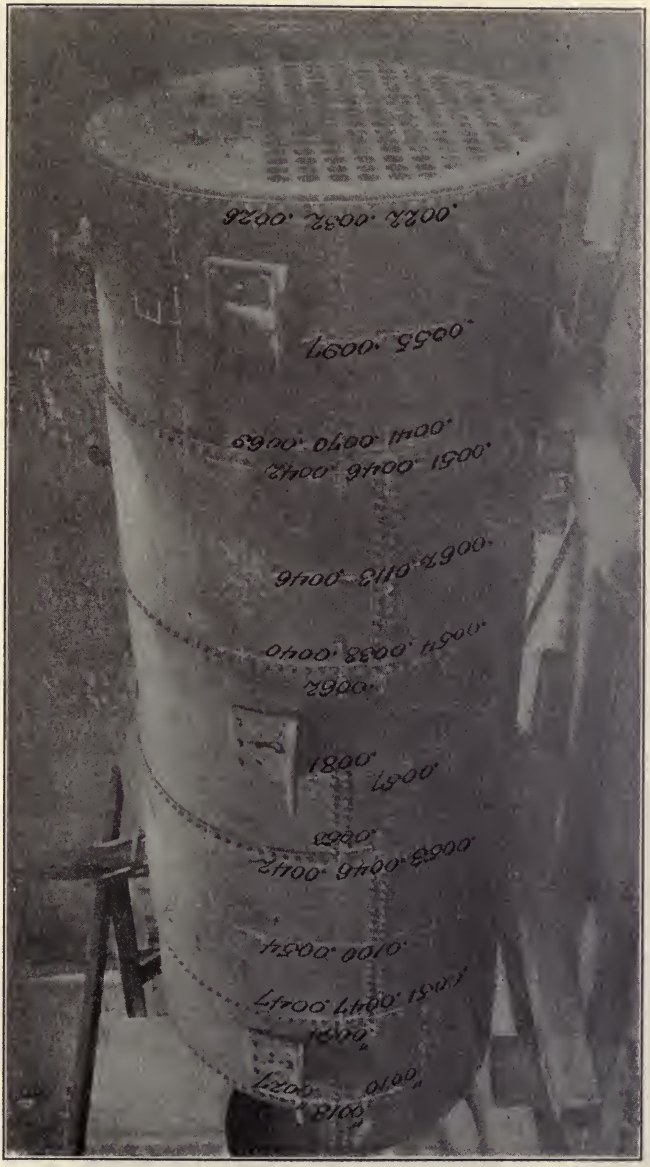


FIG. 35 TANGENTIAL EXTENSION, AT 180 LB. PRESSURE. THE FIGURES GIVEN REPRESENT STRAINS DEVELOPED ON THE LEFT SIDE OF THE BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0058 IN.

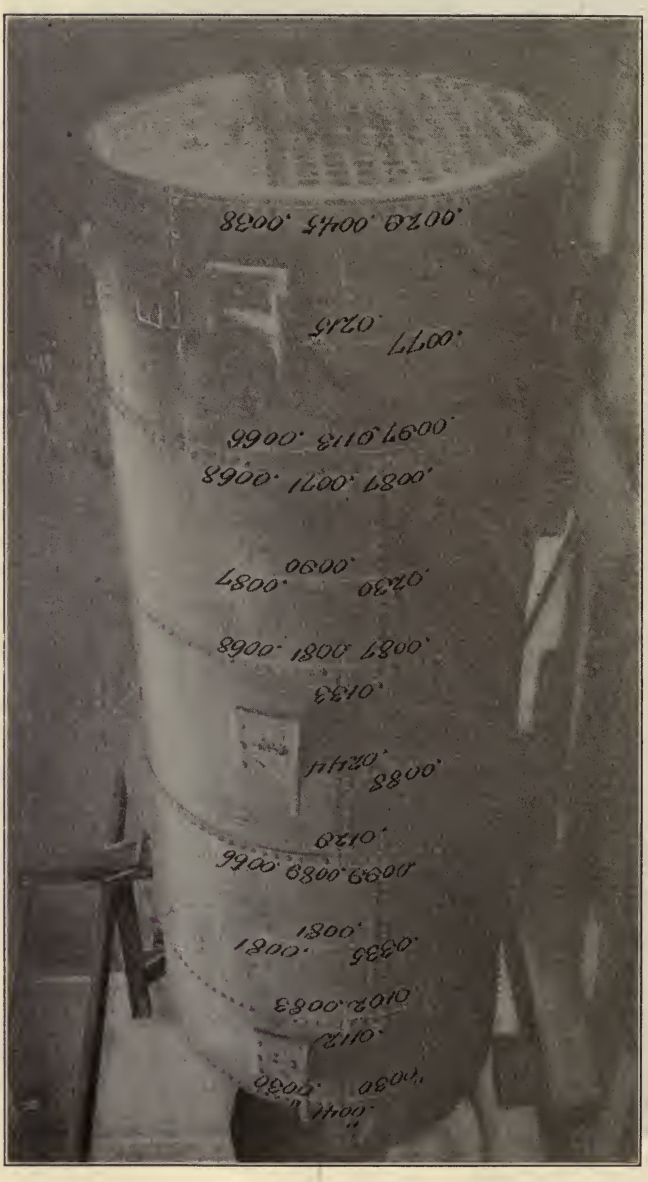


FIG. 36 TANGENTIAL EXTENSION AT 270 LB. PRESSURE. RIGHT SIDE OF BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0086 IN.

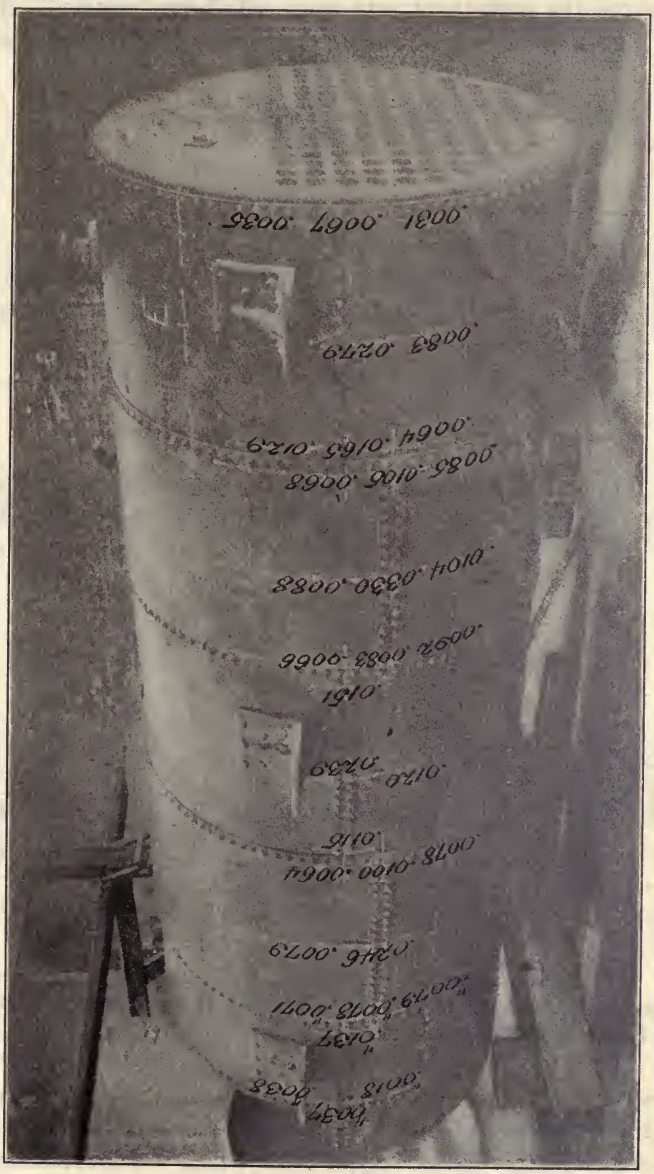


FIG. 37 TANGENTIAL EXTENSION AT 270 LB. PRESSURE. THE FIGURES GIVEN REPRESENT STRAINS DEVELOPED ON THE LEFT SIDE OF THE BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0086 IN.

56 The results of the diagrams, Figs. 18–27, refer to the total extensions of the gaged lengths, that is, they include the elastic extensions and the permanent sets when sets have occurred. The curves in Fig. 26 were plotted for the purpose of showing the elastic

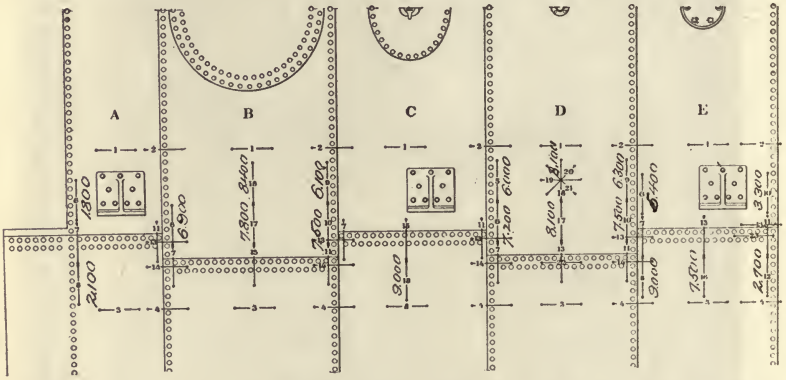


FIG. 38 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 90 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. RIGHT SIDE OF BOILER. NORMAL COMPUTED STRESS, 8640 LB. PER SQ. IN.

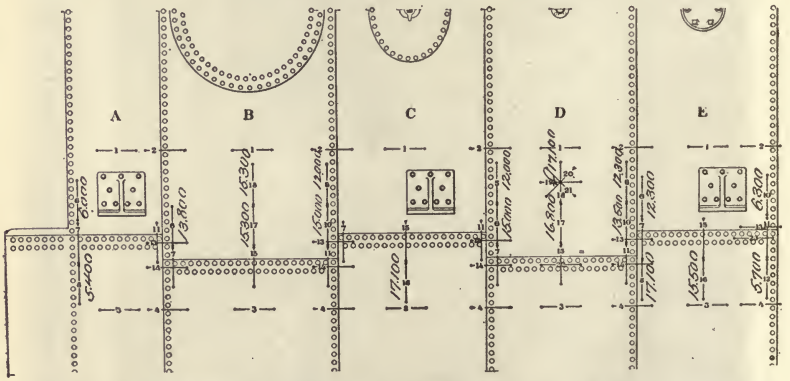


FIG. 39 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 180 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. RIGHT SIDE OF BOILER. NORMAL COMPUTED STRESS, 17,280 LB. PER SQ. IN.

extensions only, or what is equivalent to the same, the resilience of the shell taken across the longitudinal seam of course *D*, right side. The greater resilience of gaged length *D*-15, which was located across the longitudinal seam at the middle of its length, over the amount called for by the modulus of elasticity curve will be noted. This might

be taken to indicate an intensity of stress above the normal in the shell in that vicinity, or it may mean that bending and shearing stresses at the seam in addition to tensile stresses on the sheets modified the results.

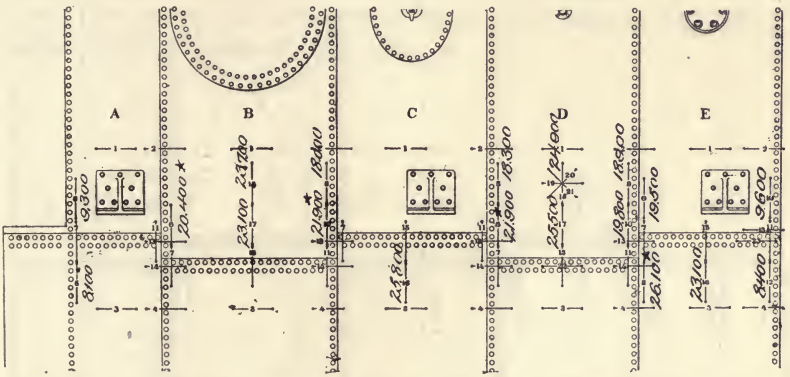


FIG. 40 TANGENTIAL STRESSES, LB. PER SQ. IN. AT 270 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. RIGHT SIDE OF BOILER. NORMAL COMPUTED STRESS, 25,920 LB. PER SQ. IN. GAGED LENGTHS ON WHICH DECIDED PERMANENT SETS OCCURRED, INDICATED BY A STAR

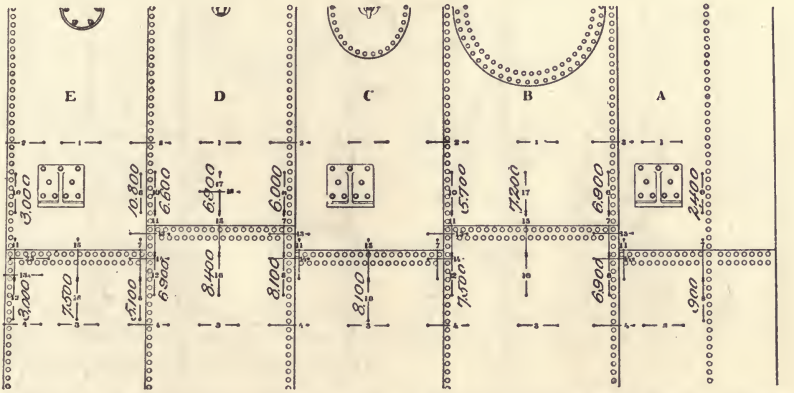


FIG. 41 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 90 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. LEFT SIDE OF BOILER. NORMAL COMPUTED STRESS, 8640 LB. PER SQ. IN.

57 The interior pressure on the boiler was increased from 300 lb., the highest indicated on the diagrams, to 335 lb., under which latter pressure rupture of the manhole patch occurred. Three of the rivets were sheared by the tangential stress of the shell, followed, apparently,

by the fracture of other rivets by tension on the stems, which pulled off the heads and finally tore the shell longitudinally along its upper element, starting this fracture at a rivet hole of the manhole opening. Fig. 28 shows the appearance of this fracture.

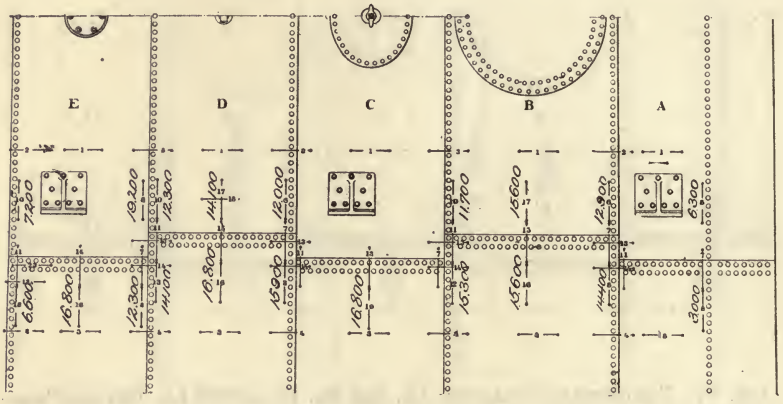


FIG. 42 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 180 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. LEFT SIDE OF BOILER. NORMAL COMPUTED STRESS, 17,280 LB. PER SQ. IN.

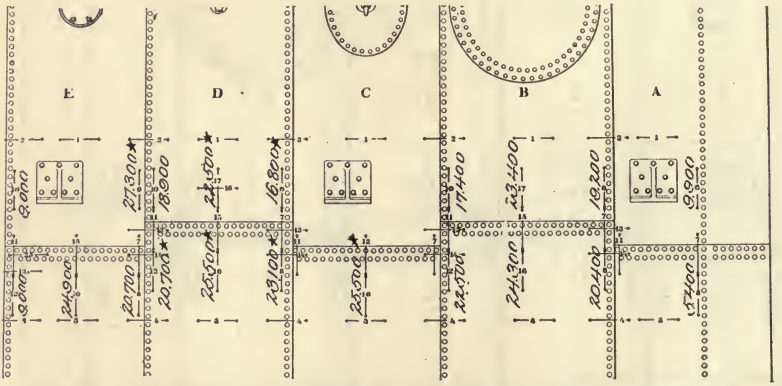


FIG. 43 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 270 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. LEFT SIDE OF BOILER. NORMAL, COMPUTED, STRESS, 25,920 LB. PER SQ. IN. GAGED LENGTHS ON WHICH DECIDED PERMANENT SETS OCCURRED, INDICATED BY A STAR

58 The shell was repaired by cutting out a portion of course *C*, across the top of the boiler and putting in a section the full length of the course and about 3 ft. wide, measured on the arc. This new

section was double-riveted to the shell at its longitudinal seams. The rivets were $\frac{13}{16}$ in. in diameter and had a pitch of 2.87 in. The rows were 1.53 in. apart, with rivets staggered, which, were, of course, hand-driven. The points of the rivets were hammered down to conical shape, low in height and with thin edges. In this respect they were less substantial than the points of the original machine-driven rivets of the seams.

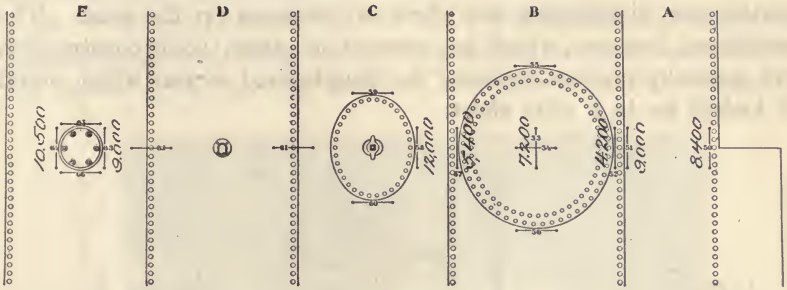


FIG. 44 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 90 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. TOP OF BOILER. NORMAL COMPUTED STRESS, 8640 LB. PER SQ. IN.

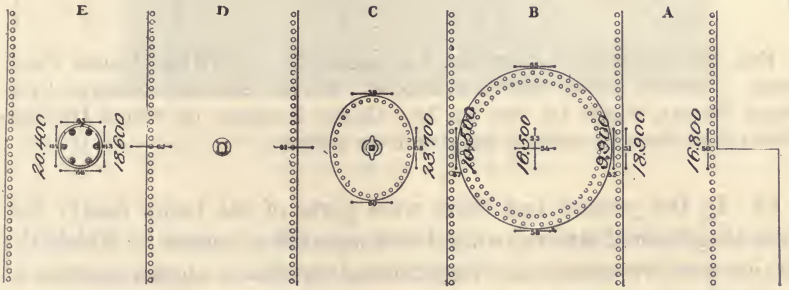


FIG. 45 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 180 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. TOP OF BOILER. NORMAL COMPUTED STRESS, 17,280 LB. PER SQ. IN.

59 The hand-driven rivets would not be expected to hold the calking as well as the machine-driven rivets by reason of the difference in their points, and the test showed that such weakness was the case.

60 The diagram, Fig. 27, shows the behavior of the seams of this new section of course *C*. The flatness of the curves indicate how early these new seams began to display rapid extension, and with so decided a movement the calking was soon disturbed and copious leaks started.

At the time of presenting these notes no higher pressure has been reached than the rupturing one of 335 lb. previously mentioned.

61 The strain measurements thus far described were those which were observed on tangential gaged lengths. In addition to this as indicated in the diagrams, Figs. 15, 16 and 17, there were longitudinal gaged lengths laid off on the shell and measured.

62 In a plain cylindrical shell the tangential extension of the metal would necessarily be attended with a definite amount of longitudinal contraction, eliminating the effect of pressures on the head. The conditions, however, which are present in steam boiler construction will generally prevent realizing the longitudinal strains which would be looked for in a plain sheet.

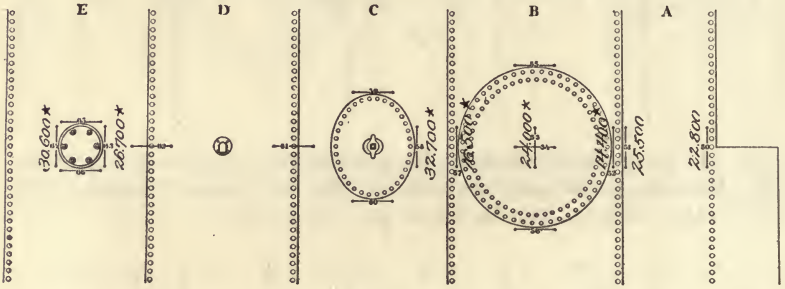


FIG. 46 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 270 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. TOP OF BOILER. NORMAL COMPUTED STRESS, 25,920 LB. PER SQ. IN. GAGED LENGTHS ON WHICH DECIDED PERMANENT SETS OCCURRED, INDICATED BY A STAR

63 In the present test there were parts of the boiler nearly free from longitudinal strains, while there were other places in which the strains were reversed, and longitudinal extension shown instead of longitudinal contraction.

64 In order to determine whether the action immediately at the girth seams was represented by the 10-in. gaged lengths which spanned them symmetrically, other gaged lengths were established on the shell, not indicated on the diagrams herewith presented. These were in pairs, one being wholly on the solid sheet, the other just stepping on to the adjacent course. The observations on these gaged lengths lead to the same results, however, as found on those which symmetrically spanned the seam.

65 The results of these observations showed that along the lower quarter of the boiler the longitudinal strains were contractions;

while along the upper quarter they were in part contractions, and in part extensions. The strains observed at 270 lb. pressure are entered on two lightly printed photographs, Figs. 29 and 30. In Fig. 29 are shown the strains which were measured on gaged lengths 1 and 3, taken on the solid sheets at the middle of the lengths of the courses. In Fig. 30 are shown the strains which were measured on gaged lengths 2 and 4, taken across the girth seams. Minus signs before the figures indicate contractions while plus signs indicate extensions.

66 It will be observed that the lower part of the shell contracted longitudinally, notwithstanding the fact that the tubes were extended by reason of the exterior pressures to which they were subjected.

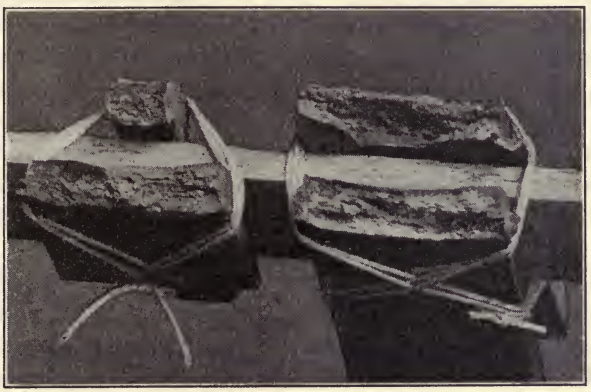


FIG. 47 LAMELLAR APPEARANCE OF METAL, SHEET *C*, FRACTURED BY BENDING WHILE AT A BLUE HEAT

67 This behavior calls for bending at the flanges of the heads to compensate for the difference in direction of these movements. The six through braces would relieve the shell of a portion of the longitudinal tension coming from the heads in the upper half of the boiler.

68 Longitudinal gaged lengths on the upper part of the shell showed diminished contractions over those observed on the lower portion, or displayed strains of extension. On the very top of the boiler the strains were extensions of a pronounced order.

69 It was found on diagonal gaged lengths, laid off on courses *B* and *D*, upper quarter of the boiler, that greater extensions were displayed on the converging diagonals over those of diverging directions. The converging diagonals, at 270 lb. pressure, extended 0.0047 in. and 0.0042 in. respectively, against 0.0036 in. and 0.0028 in. displayed on the diverging gaged lengths.

70 The location of the diagonal gaged lengths on course *D* are shown in Fig. 14, similar lengths having been laid off on course *B*. These results are entered in Fig. 30, in addition to the longitudinal strains. The longitudinal strains, all being those of extension, observed on the top of the boiler, have been entered on the diagram, Fig. 31. The strains for each pressure, 90, 180, and 270 lb. respectively, are given. The range and variability of these measurements are seen to be very pronounced.

71 A series of lightly printed photographs, Figs. 32 to 37 inclusive, are presented, on which are entered the measured tangential strains observed at pressures of 90, 180 and 270 lb. respectively.

72 The strains observed on both the right and the left sides are entered, however, on prints representing the right side of the boiler. It will be understood that the longitudinal seams of the left side were the reverse of those on the right side as regards their respective heights. That is, on the left side the seams of course *B* and *D*, were three rivets above instead of three rivets below the others, as shown in illustrations of the right side. The tangential strains called for by computation based on a 30,000,000 modulus of elasticity are stated for each pressure in the captions of the figures.

73 The range in measured strains above and below the computed amount will be noted upon inspection of Figs. 32–37. The strains were least in amount at the heads, and generally greater rigidity was displayed at the intermediate girth seams than in the solid sheets, under the earlier pressures. At the middle of the length of the longitudinal seams the maximum extensions were developed, witnessed in this examination of the behavior of the shell.

74 The results thus far presented, with the exception of those on Fig. 26, have included both the elastic strains and the permanent sets of the different measured lengths.

75 The permanent sets have been subtracted from the extensions and the stresses corresponding to these resiliences computed, and the results entered on another series of diagrams, Figs. 38 to 46 inclusive. These results show the tangential stresses, in pounds per square inch, found in different parts of the solid sheets of the shell when the boiler was subjected to pressures of 90, 180 and 270 lb., respectively.

76 In looking over those results the usual influence of longitudinal seams, such as were used in this boiler, in intensifying the tangential stresses in the adjacent solid sheets, may be pointed out. The excessive stress at the side of the single-riveted manhole patch is

clearly shown in the results. The high stresses at the sides of the safety-valve nozzle under the maximum pressure will also be noted.

77 While the results are consistent, nevertheless as an engineering structure the distribution of stresses exhibits a range far beyond that which is expected in other classes of constructive work. The type of boiler being one of the simplest, the extension of this method of test to other types would seem desirable. Such tests might assist in establishing satisfactory rules for steam boiler construction and might reasonably be expected to aid in the framing of regulations governing allowable pressures.

78 Additional tests were carried out, in which the effect of changes in the manner of supporting the boiler was inquired into. It was supported on the four end lugs in one test, and again in another test most of the weight was carried by the middle lugs. In each case there was a modification in the measured strains, although not in a marked degree. At the end of the test the several courses were seen to have been visibly extended in diameter between the girth seams.

79 The chemical composition of the steel in course *C* was as follows:

Carbon	0.22
Manganese	0.43
Silicon	0.046
Sulphur	0.028
Phosphorous	0.043

It is recalled that this particular brand of steel at the time of its manufacture was not infrequently found to possess a decidedly laminated structure. The laminations were not large, nor likely to cause blisters in the boiler, but they were in places quite numerous.

80 The metal from course *C*, the only sheet yet examined, was found to have a laminated structure. Fig. 47 shows the appearance of some fractured strips from this course, which were bent while at a blue heat, in order to develop the lamination of the plate in a pronounced manner. The metal drifts well, a $\frac{3}{4}$ in. diameter punched hole having been drifted cold to $1\frac{1}{4}$ in. diameter without rupture.

81 The services of Mr. P. W. Brunner, and J. W. Herrity are acknowledged, whose skill as manipulators is shown by the internal evidence of reliability which these measurements, taken by them, furnish.

DISCUSSION

FRANCIS B. ALLEN. The paper contains some very interesting data which are more complete than any we have ever had so far as I know, in tests of boilers of this description. In my own experience, it is a very difficult problem to convince a manufacturer when a battery of boilers has given good service and, superficially at least, is without patches or other evidence or deterioration, that it is necessary to reduce pressure in order to maintain so far as possible the full factor of safety. I have in mind a set of boilers which was in operation for 27 years, in which the original pressure allowed was 85 lb. By successive cuts as the boilers increased in age, this pressure was reduced to 60 lb. and upon inspection a recommendation was made that the pressure be reduced to 50 lb., or preferably that newer and stronger boilers be submitted. This meant an expense of thousands of dollars and the manufacturers naturally wished to use the boilers a little longer and so make a greater return upon the investment.

In the belief, however, that it would not be advantageous to take the risk involved, they were taken out; and the test which the paper describes established the fact that while there was an ample factor of safety according to the requirements of law and custom, the recommendation to remove the boilers was fully justified because they had approached the danger line as closely as should be allowed. The fact that these boilers described in the paper were exceptionally good and had been carried by one organization and insured by another from the time they were put into commission until they were taken out, makes the record and data unusually valuable.

THE AUTHOR. The two boilers under consideration present features of interest because of the long term of their service and their excellent condition when the tests were made. Originally, these boilers were run under 90 lb. steam pressure which was eventually reduced to 60 lb., the pressure carried at the time they were taken out of service.

An inspection of the interior of the first boiler showed little or no scale on the sheets, excepting a limited amount of loose material on the bottom of the boiler. The upper row and side vertical rows of the tubes were free from deposit, while the interior ones had a deposit on the rear quarter or half and at their immediate front ends. The tubes, however, were free from pitting and general corrosion.

The condition of the boilers was such that the results of the strain measurements may be taken as representative of the behavior of boilers of their type in respect to the distribution of stresses in different parts of the shell when under pressure.

In the test of the first boiler, such copious leakage at the single riveted longitudinal seam of the dome occurred at 266 lb. pressure that its removal was necessitated. At that time, the metal at the base of the dome was overstrained and lines of scale were detached, showing that the elastic limit of the metal had been passed. The manhole frame was the next to yield, fracturing at 270 lb. pressure. Several of the stays at the front head at 295 lb. pressure ruptured. The test was then discontinued and the boiler dismantled. The metal from the shell and some of the tubes was reserved for subsequent examination and test.

Prior to testing the second boiler it was divested of its dome and manhole frame and the heads were strengthened by means of six through-braces. With this boiler a rupturing pressure was reached at 335 lb., the rupture occurring at the single-riveted patch covering the manhole opening. A section of this course was cut out and the shell repaired with a new piece of plate, which was double-riveted. Hand-holes necessary for riveting were made in the new plate. No higher pressure has yet been reached than the earlier one of 335 lb., leakage overcoming the pumps and taking place at the longitudinal seams and particularly at the new hand-riveted seams.[1]

Concerning the distribution of the stresses in the shell, at few of the measured lengths was the tangential extension of the shell found to correspond closely to the computed extension, although there were places where there was full agreement. Generally, the tangential extensions in the vicinity of the girth seams were less than the computed values. Naturally the edges of the end courses over the heads displayed only a limited amount of extension.

Across the longitudinal seams, taken at the middle of the lengths of

[1]A pressure of 370 lb. has since been reached. The sheets show further deformation but have not yet ruptured.

the several courses, the extensions were greater than called for by computation for the solid sheets. This increased yielding at the longitudinal seams became more pronounced as higher pressures were successively applied, the extensions in the vicinity of the girth seams, however, remaining less than the normal amount.

A place characterized by unusual rigidity was found on the top of the boiler in the immediate vicinity of the double-riveted manhole patch and the girth seam, while in the next course at the side of the manhole patch unusual extension was displayed.

The relative rate of extension observed under early pressures was maintained without material change under the subsequent higher pressures.

The longitudinal strains were found to be contractions on the lower part of the shell, in part contractions and in part extensions on the upper part of the shell, while on the top of the boiler they were all strains of extension.

The method of measured strains illustrated in these results is a convenient one for extending the scope of research work beyond the usual limits of laboratory tests as conducted in testing machines. Knowledge pertaining to the distribution of stresses, important in structures of all kinds, may be ascertained in this manner. The constructive merits of different types of boilers admit of being ascertained by the method of measured strains.

It would seem feasible, judging from the indications of the present tests, to determine the merits of different types by means of observations confined within the limits of the usual hydrostatic test pressures, as applied to new boilers. That is, an adequate opportunity is presented in the hydrostatic test to acquire this information and in no wise impair the strength of the shell or other parts by reason of having made such strain measurements.

The obvious value of such information will not require further mention. It is believed that herein is presented a method of test which will lend material aid in furnishing a rational basis on which to establish rules regarding allowable steam pressures.

With the present knowledge available on the effects of repeated stresses in destroying the integrity of structures, it is apparent that information pertaining to the behavior of structures under working loads is desirable. Most failures emphasize the need of more complete data on working conditions rather than any deficiency of knowledge on the ultimate strength of the component parts taken individually.

SYMPOSIUM ON STEEL PASSENGER CAR DESIGN

No. 1388 *a*

INTRODUCTION

By H. H. Vaughan, Montreal, Canada

Member of the Society

The advent of the steel passenger car has brought with it many new problems and an opportunity for more diverse opinions than any other change that has taken place in car equipment. The construction of the wooden passenger car developed along fairly uniform lines. The varieties of framing were few and the differences unimportant, while the introduction of steel platforms, wide and narrow vestibules, reinforced end and sill construction and similar improvements occurred gradually, and with practically similar designs on all railroads. The change from wood to steel in freight car construction resulted in the abandonment of designs that had almost become standardized and the introduction of many new types, but in this case the principal problem, other than that of obtaining satisfactory designs, has been the extent to which it was advisable to use composite or all-steel construction.

2 In the case of the passenger car, the types to be employed will probably not be changed by the substitution of steel for wood. The increase in capacity that has taken place in freight equipment cannot be duplicated in passenger cars, and there appears to be no tendency at present toward any increase in length or carrying capacity. The questions that now confront us relate rather to the design and construction of cars of the present type and of the materials that may be advantageously employed in place of the wood which has been used for so long. They are complicated by the necessity of providing for greater safety for the passengers than was secured in the wooden car, with an equal degree of comfort and the difficulty of anticipating the behavior of this new equipment in the case of accident. Certain difficulties such as the best systems for heating, lighting and ventila-

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

tion, are common to both steel and wood construction, and improvements in these matters pertain to general progress rather than the use of steel construction. The following list, while probably incomplete, outlines in a brief way the important variations that must be considered in deciding on the preferable construction of steel passenger equipment:

Framing	Steel underframe	
	All-steel frame.....	Center girder Side girder
Outside finish.....	Plated Sheathed	
Roof construction.....	Clear story Circular	
Inside finish.....	Steel Wood	
End construction.....	Design and strength	
Floor	Design and material	
Insulation	Material	

No doubt questions of equal importance have been omitted, and in many cases those mentioned require careful consideration with regard to degree, as for instance, the strength of the framing or the thickness of the insulation. The list illustrates, however, the diversity of possible solutions of the preferable steel passenger car, and the following personal opinions are presented for the purpose of opening the discussion:

3 The steel underframe does not appear to be a satisfactory or permanent development. There is but little saving either in weight or cost over the all-steel construction, and it is difficult to see how the same strength in case of accident can be obtained. Experience will show whether the wood superstructure can be secured in such a way as to prevent working as the car gets old, but as it cannot be arranged to carry any weight this appears questionable. It can hardly be regarded except as an intermediate step between all-wood and all-steel construction.

4 In all-steel construction the side-girder car presents advantages, but as in freight construction, both types will probably persist. The side-girder construction obtains greater strength on the side framing without superfluous weight, and it is possible that greater framing strength may prove necessary. With equal strength of side framing the side-girder car may be made lighter than the center-girder type, and the weight of steel passenger cars is one of the most serious problems to be faced by any railroad not having a level line. Ameri-

can passenger equipment was already excessively heavy per passenger carried with wood construction, and the use of steel has increased this weight from 10 per cent to 20 per cent, which is a most serious matter. Apparently side-girder cars as so far constructed have a decided advantage over the center-girder type in their light weight and greater strength in case of accident tending to crush in the side of the car. This will probably lead to the use of this type on roads on which weight is of importance.

5 In spite of the many advantages of the sheathed car in case of construction and maintenance, it appears that the cost and weight of the additional metal will prevent its extensive use. This question is chiefly one of appearance and convenience, and is of minor importance.

6 The circular roof has been extensively introduced on steel passenger cars on account of its lightness and simplicity of construction. It has the objection that deck sash ventilation cannot be employed. The Pullman Company while using the clear-story roof have, however, discontinued the use of deck sash ventilation, so that evidently in their opinion this objection is not important. The deck sash is, however, of value in a standing car, and when properly screened is certainly advisable in hot weather, especially when the road is dusty. The Canadian Pacific Railway have compromised on this question and are using a roof of approximately circular form with deck sash. The strength and simplicity of the circular roof is retained with the ventilating qualities of the clear story type.

7 The preferable material for inside finish is a matter for future decision. With the ample protection afforded by a steel car against accident, there does not appear to be any objection to wood inside finish on the ground of safety. It is more ornamental than steel and a better insulator. Probably on no question in passenger car design is opinion so divided amongst both railroad and car builders. There is today very little difference in cost, and it certainly appears probable that in the future the tendency will be to adopt steel interior finish if not entirely, at any rate to a great extent.

8 The construction of the ends of the cars has received considerable attention, and the strength now usually employed is enormously greater than anything attempted in wood construction. Several excellent designs have been devised, which will probably be referred to in another paper.

9 The floor construction in steel cars is entirely different from that in wooden cars, and is usually of metal covered with a flexible

cement. In constructing a sample car for the Canadian Pacific Railway the writer used in addition an underfloor covered with insulating material, and covered the cement with ½ in. of cork. This car was also exceptionally well insulated at the sides, 2 in. of cork being used next the outside plating. Tests during the past winter have shown that this car is actually warmer than the ordinary wooden car, the same amount of heating surface being used in both types. The floor was tested by taking the temperature of water standing in cans on the floor, there being no practical difference between the results in the wood and steel cars. The question of insulation is an important one, both in hot and cold weather, and while other insulation might no doubt be equally effective, it is interesting to be able to advise that with proper insulation there is no question of the steel car being satisfactory.

No. 1388 *b*

PROBLEMS OF STEEL PASSENGER CAR DESIGN

By W. F. Kiesel, Jr., Altoona, Pa.

Member of the Society

Whenever it becomes necessary to adopt a policy representing a complete departure from existing policies involving a new theoretical structure from foundation up, many problems, some entirely new, have to be solved. The increasing cost of lumber, the desire for longer and stronger cars, and other considerations indicated the desirability of making a determined effort to develop a satisfactory steel passenger car. The object of this paper is to review a few of the problems encountered, beginning with:

2 *First: Can we afford it, and what will it cost, compared with wooden cars?* Tentative designs were prepared and carefully analyzed by a committee of representatives of carbuilders and railroads. The summary of their report was that at first steel passenger cars would cost approximately 20 per cent more per passenger than wooden cars of the best existing types, but that the steel cars would probably cost much less to maintain. They also reported that on account of the increasing cost of good lumber, and the probable decreasing cost of manufacturing steel cars, not many years would elapse before the cost of steel cars would be no more than, if as much as, wooden cars. Those who have been in close touch with the development of the steel-car industry know that at the present time steel cars cost no more than equivalent wood cars.

3 *Second: Shall the cars be all steel, or steel frame with wood lining?* Differences of opinion still exist on this point. Both types of car have been built, and each has strong advocates.

4 In the all-steel car the steel lining can be securely riveted to the framing and adds somewhat to the strength of the complete structure, but as steel is a good conductor it carries away the heat of a body coming in contact with it, and, therefore, will always feel cold, even when the temperature in the car is sufficiently high. Satisfactory results have been realized from the use of a double steel lining

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between seats, forming a hot-air duct, extending from the heater pipes to the window sill, with outlet through small holes in the lining proper, located immediately below the window sill in the lining proper.

5 Wood lining requires considerable wood furring, and adds weight to the car without adding to the strength. As the steel frame of a long passenger car may vary as much as ½ in. between extremes of temperature, it is necessary to make allowance in the construction of the wood lining for this variation in length. As a car with metal lining riveted to the framing has the advantage in strength, weight, and cost, it will gain in favor; in fact, it would be at present universally preferred if all railroad shops had practical experience with steel lining, and the necessary proficiency and machinery for its manufacture.

6 *Third: Insulation.* Three general principles have been used for car insulation: (*a*) Wood lining; (*b*) by placing insulating material on the outside of steel lining; (*c*) by placing insulating material on the outside of the steel lining, and on the inside of the steel sheathing.

7 Experiments have been made also with other methods, such as completely filling the space between sheathing and lining with block magnesia and magnesia cement. The problem that presents itself is: Given a car body with a comparatively smooth exterior surface protected by several coats of paint, double walls, painted on both sides—if of steel, isolated air spaces, rather large in volume, between the walls, an inside cubic volume in which the air must be continually renewed, and a window surface of about one-third of the area of the side walls. When single windows are used the air close to the windows is cold in winter, and warm in summer. Double windows improve the situation materially.

8 Experiments made to determine the difference between a wooden and a steel coach, with doors and windows closed, standing on a siding exposed to the sun in hot, summer weather, showed a difference of one to two degrees in favor of the wooden coach. One day's readings showed an average of one degree difference in temperature in favor of the steel coach, which had insulation only on the outside of the lining. The results of several years' experience indicate that the lining must be insulated throughout, and, if the spaces between lining and sheathing are properly isolated, little is gained by insulating the sheathing, and more will be gained by the use of double windows. Furthermore, the heat lost in cold weather by conduction through and radiation from the walls, in cars with insula-

tion on the lining alone, is negligible when compared with the heat carried off by adequate ventilation.

9 *Fourth: Protection and safety of passengers.* This problem involves providing adequate strength for carrying the load, also to prevent collapse or crushing in wrecks, and efficient brakes.

10 The laws governing load-carrying strength are well known, but this cannot be said of the laws governing wrecks. Each wreck forms a separate study, and we seldom find two that can be placed in the same class. The study of wrecks, which, unfortunately, do occur, shows that the car underframe must be reasonably strong to resist end strains, that the ends of the superstructure must be reinforced with strong vertical members, and that the car must not collapse when rolled down an embankment. The gradual elimination of crossings at grade has materially decreased the danger of strains directed against the sides of the car.

11 Early experience with steel freight cars showed clearly that the men handling cars in yards believed that all cars built of steel could withstand much rougher handling than wooden cars. Although the resultant damage to both kinds of freight cars had its disadvantages, it developed a better knowledge of the relative value of steel and wood in car construction, led the designer to abandon the basis of ultimate strength of the material, and to substitute the basis of elastic limit, and finally to select a ratio of 4 to 1 as the relation of the elastic limit of steel as used in cars to that of good timber.

12 That not all designers of steel passenger cars had the advantage of this knowledge, or profited by this experience, is evidenced by some of the car designs which have been illustrated in the technical papers in the past years and which proved fundamentally defective.

13 Selecting from the last generation of wooden cars one used in heavy trunk line service, with four 5-in. by 9-in. wooden sills bunched together near the center, and so located as to be nearly uniformly affected by the end strains, steel platforms with draft gear securely attached, and the remainder of the car to correspond, the analysis of its end-shock resisting capacity leads to the consideration of the elasticity of the material, the transverse bracing preventing buckling, the concentration of strength near the longitudinal center line of car, and the reinforcement at the platforms.

14 The wooden car, therefore, meets many of the requirements enumerated before. A corresponding steel car should have a center sill area of 45 sq. in. braced against buckling, a strong and efficient draft gear as a substitute for the elasticity of the wood, and a ratio of 0.04

for stress to end force, the calculations to include consideration of lever arm of force below neutral axis of the center sills. For lighter service a steel car with center sill area of 32 sq. in. and a ratio of 0.05 for stress to end force may be considered as a substitute for a wooden car with four 4-in. by 8-in. sills bunched near the center of the car. The use of steel permits a distribution of material to better advantage than is possible with wood. The box girder center construction is continually gaining in popularity, the strong vertical members at car ends, to prevent one car overriding and penetrating the superstructure of another car, are now considered a necessity, and a superstructure, including roof sufficiently strong to bear the car when turned upside down without collapsing, is very desirable.

15 To avoid making this paper too long other interesting problems will be omitted, but the truck problem deserves brief consideration. There are four-wheel and six-wheel trucks. They have 4¼-in. by 8-in., 5-in. by 9-in., 5½-in. by 9-in. and 5½-in. by 10-in. journals.

16 The impression that cars with six-wheel trucks necessarily have better riding qualities than those with four-wheel trucks has proved to be incorrect. The substitution of four-wheel trucks for six-wheel trucks saves about 18,000 lb. per car. Increased journal bearing surface obtained by an increase of diameter of journal only is of little or no benefit in preventing hot boxes, because the periphery velocity increases in the ratio of the diameters. The weight per journal should not exceed 1500 lb. per in. length. A long spring base, low-lying center plate, and anchoring the dead levers to the car body instead of to the truck frame promote smooth action and easy riding at all times. The equalizing springs should, therefore, be placed as near to the journal boxes as possible, or directly over the boxes, and the bolster springs should be on or near the center line of truck sides. If the dead levers of the truck brake are anchored to the car body, the truck frames have no tendency to tip up when the brakes are applied, and the jarring effect is entirely eliminated. A special axle with 5½-in. by 11-in. journal for passenger cars would be of material benefit, would permit using four-wheel trucks under all coaches and 60-ft. baggage cars, and longer cars with six-wheel trucks would have sufficient margin for the excessive loads sometimes encountered and the danger of hot boxes would be avoided.

No. 1388 *c*

UNDERFRAMES FOR STEEL PASSENGER CARS

By John McE. Ames, New York

Member of the Society

This paper will be confined to underframes of steel passenger cars for through service, or those at least 70 ft. long, and will not attempt to discuss those of suburban or individual service, whose underframes are not subjected to the same severe service strains.

2 The underframe is called upon to perform several functions. Not only must it sustain the weight of the superstructure and load, but withstand impact, oscillation and pulling strains without distortion. Were it not for these conditions the underframe might be considered as a bridge resting upon the center plates and side bearings as piers. Were we to design to meet only the carrying requirements the problem would not be difficult, but the design must also be commercial, not over heavy and in addition sufficiently strong to resist impact; commercial in that plates and shapes employed are such as may readily be secured from the steel mills, and not so heavy as to bring undue work upon either the hauling locomotive, rails, frogs, bridges, etc.

3 The natural division of such designs is:

a Underframes designed to carry equally on all sills

b Underframes designed to carry on center sills only

c Underframes designed to carry on sides only

c Underframes designed to carry on sides and center sills

4 Each of these types has its partisans and each type is in successful operation today. The first is the type used abroad almost universally and at home for repairs under wooden cars, the bodies of which are too good to destroy but need better underframing. With most of the foreign cars the body rests upon and is bolted to the underframe from which it may readily be removed. The buffing and draft conditions differ from ours in that the buff is taken through

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the side sills by the use of separate side buffers, and the draft through the center sills thus permitting a distribution of metal in each sill member that may produce uniform stress.

5 An example of the first type designed for a wooden superstructure, consists of four deep sills of what is known as the "fish-belly" type (Fig. 1). These center sills are composed of 5/16-in. plates, 30 in. deep at the center with 3 in. by 3 in. by 3/8 in. angles riveted along the top and bottom edges; the plates reduced to a depth

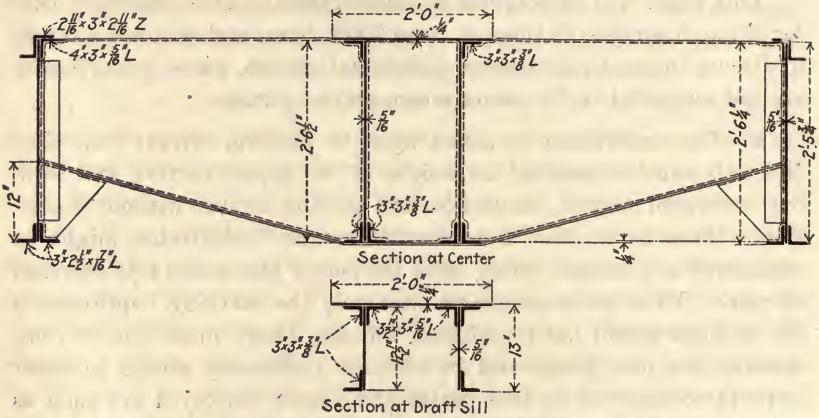

FIG. 1 TYPE *a*: WEIGHT OF CAR CARRIED EQUALLY ON CENTER AND SIDE SILLS

of 12¾ in. over the bolster. The center sills have a square inch section of 37 at the center, just as the side sills, and 26 at the draw gear. One disadvantage in these long plate sills is that when punching the line of holes along the edges the plate becomes distorted and wavy. It is then difficult to rivet the angles in place and obtain their full value. Again, in case of accident and the dropping of the underframe upon the roadway, the bottom angles are bent or broken, making a difficult repair operation.

6 In general the deep side sill has been discarded because of the difficulty of inspection beneath the car. The deep center sill is much in vogue at present because it looks strong, but on a car with deep center sills inspection must be made of the parts attached to the underframe from one side of the car at a time, and the introduction of axle light equipment becomes difficult on account of the interference with the deep sills. Again, to sustain its own weight without deflection on a 60 ft. span, too much weight of metal is required to make such a sill economical.

7 Of the second type, that is, with the whole weight to be carried on the center sills, a common form (Fig. 2) has center sills of two special 18-in. channels with ½-in. cover plates top and bottom, all sections extending full length of the car in one piece. The box girder so formed has a square inch section of 50, and the superstructure load is transferred to these sills by means of four cross bearers, two of which take the place of the body end sills in other design. There are no side sills as such, the angles here shown simply forming the attachment for the superstructure. The parts are usually assembled with the bottom of the sills upward and allowed to deflect. The girder is then reversed and the camber straightens out by the weight of the

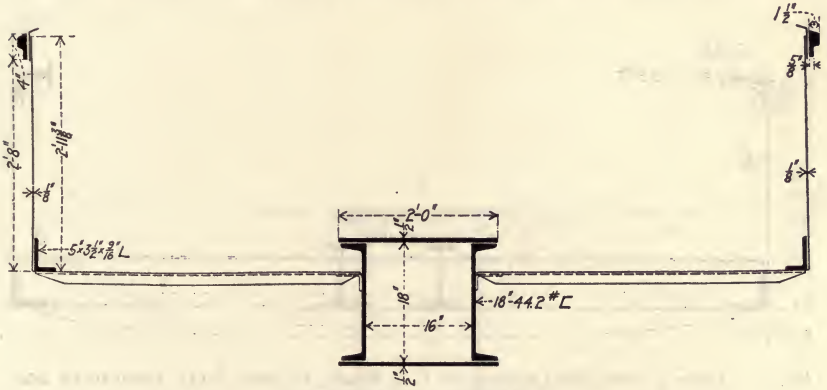

FIG. 2 TYPE *b*: WHOLE WEIGHT CARRIED ON CENTER SILLS

metal. The sills are the same depth and section throughout their entire length and with this construction a truck of special design must be used, the center plate of which must be nearer the rail than usual. The weight of the body rests upon the side bearings as well as the center plate. About 20 sq. in. of metal in the sides is available to help sustain the load. The service given by this underframe has been excellent.

8 The third type, with all the weight carried by the car sides has the center sills used only for buffing and pulling. An example shown in Fig. 3 has two I-beams running full length of the car in one piece, with a square inch sectional area of 23. They are held up by the three cross bearers which pass under and are attached to them. There are no side sills, the carrying members being the sides of the car. These members are composed of ⅛-in. plates, about 36 in. deep,

stiffened vertically by the window posts and having a 6 in. by 6 in. by $\frac{5}{8}$ in. angle at the bottom and an equal square inch section of metal at the belt rail, the two girders having a square inch section of 48 in all. With this construction a substantial body bolster is essential, as the load must be carried at the bolster extremities. Usually a cast-steel structure, built into the underframe and securely riveted to it, is used, the metal may thus be economically distributed. With an underframe of this type there is no trouble due to difficulty of inspection or interference with attachment for axle light or other equipment under the car.

9 The fourth type (Fig. 4) is a combination of types *b* and *c*. Here deep center sills are used, having a square inch section of, say

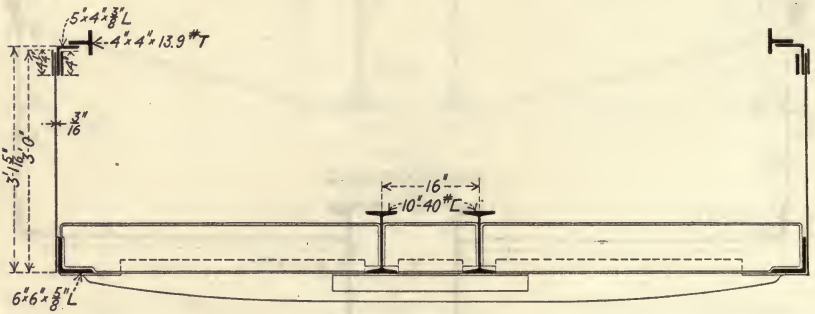

FIG. 3 TYPE *c*: WEIGHT CARRIED BY CAR SIDES, CENTER SILLS USED ONLY FOR BUFFING AND PULLING

40 at the center and 39 in cast steel at the draw gear. The side girders have a square inch section of 21 in the two. Most underframes of this type now in service are built with cast-steel end portions which include in one casting the body bolster, platform, side and center sills extending as far back of the bolster as may be necessary to secure a substantial connection to the center sills proper. This center member we do not consider as properly constructed for the reason that the section is unbalanced, an excess of metal being used on the top. Heavier angles or a cover plate should be used on the bottom, which would add about 10 sq. in. or more of metal.

10 The four types illustrated are of underframes actually in service. A comparison of cross-sections discloses the fact that no matter from what angle the designer has approached the problem, approximately the same square inch cross-section has resulted. If, therefore, any one type has an advantage in weight over the others,

it must be attributed to difference in the cross members of the underframe.

11 These four prevalent types have been recognized by the United States Government. The specifications of the Postoffice Department for the construction of steel postal cars provide as follows:

a Heavy center sill construction, the center sills acting as the main carrying member.

b Side-carrying construction, the sides of the car acting as the main carrying members, having their support at the bolsters.

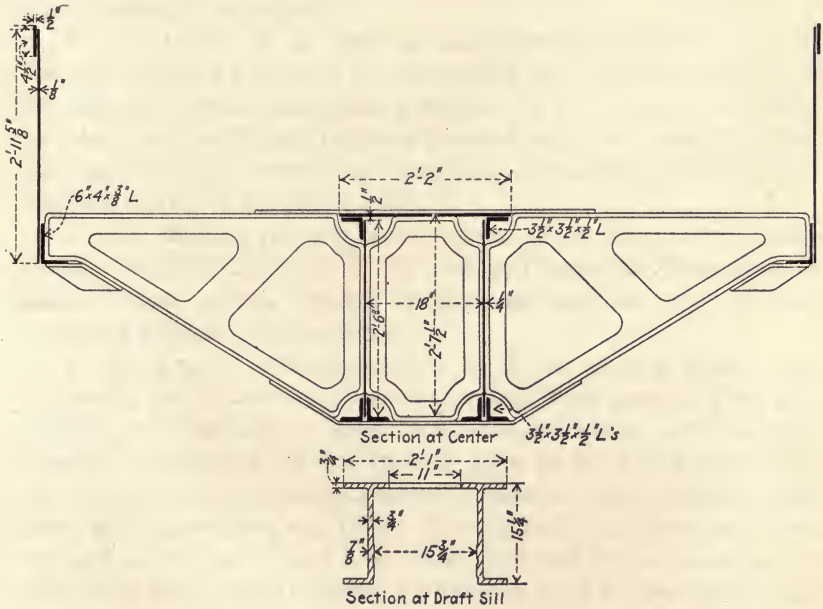

FIG. 4 TYPE *d*: WEIGHT CARRIED BOTH BY CENTER SILLS AND CAR SIDES

c Underframe construction, in which the load is carried by all the longitudinal members of the lower frame. The superstructure shall be of steel.

d Combination construction in which the side frames carry a part of the load, transferring it to the center sills at points remote from the center plate for the purpose of utilizing uniform center sill area.

12 While several of these types have been in service for a number of years the required time has not passed in which to develop structural defects due to unseen causes, such as fatigue of metal, crystallization, etc. If such defects exist they should make themselves known during the next three or four years, if freight construction is any criterion.

No. 1388 *d*

ROOF STRUCTURE FOR STEEL CARS

By C. A. Seley,[1] Chicago, Ill.

Non-Member

Roofs for steel passenger equipment cars are of two classes, the clear-story type with minor variations and the oval type. As regards contour and general appearance, they are the same as the long established standards for wooden cars, but varied as to constructive detail, due to materials employed.

2 The advent of the steel car has rather encouraged the use of the oval or round roof, as it is often called, particularly for cars used for baggage, express, and postal purposes. It is cheaper to build and maintain and fulfills requirements for such cars. For passenger cars the clear-story type prevails very generally, as it assists in lighting and ventilation and in decorative effect.

3 The framing for oval roofs consists of car lines, each a single member, bent to the shape of the arch and extending from plate to plate. There are no through longitudinal members and the roof sheets are riveted to the car lines.

4 Framing of clear-story roofs is of two general classes, one employing car lines of one piece extending from plate to plate and carrying the longitudinal upper deck sills and plates, and the other class an extension of the side framing posts as far as the upper deck sill. To these extensions are attached a member which comprises deck posts and upper deck car lines. It is difficult to approximate the strength of the more direct lines of the oval roof in the design of the clear-story roof, and all riveted connections must be thoroughly considered. The deck sills and plates are through members, act as end stiffeners, and add to the longitudinal strength.

5 The shape of the car lines of either type of roof should be such as to facilitate fastening of roof and of the inner ceiling or finish, and between these there should be a generous amount of insulating material to intercept the heat of summer and the cold of winter.

6 The committee of engineers who framed the specification for

[1]Mechanical Engineer, Rock Island Lines.

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full postal car construction, which was approved by the Postoffice Department in March 1912, contains the following paragraphs in regard to the roofs of such cars and is probably as authoritative a statement as there is available. The strength of roofs of some cars that have been rolled over in accidents has been checked against the formula used, and it has been found ample to afford support against serious roof distortion in such cases.

7 The postal specification reads as follows:

ROOF

"General

The roof may be of either the clear-story or turtle-back type, depending on the standard contour of the railroad for whose service the cars are built. In the clear-story type, the deck plates shall be in the form of a continuous plate girder, extending from upper-deck eaves to deck sill, and either built up of pressed or rolled shapes or pressed in one piece from steel plates. The car lines may be either rolled or pressed steel shapes, extending in one length across car from side plate to side plate, or may extend only across upper deck. In the latter case the lower deck carlines may be formed by cantilever extensions of the side posts or by independent members of pressed or rolled shapes. In the turtle-back type, the car lines may be of either pressed or rolled shapes, extending in one length across car between side plate and side plate, or may consist of cantilever extensions of the posts.

"Car Lines

The projected area of the portion of roof in square feet, supported by car lines, divided by the sum of the section moduli of the carlines, must not be more than 100.

"Roof Sheets

Roof sheets, if of steel or iron, shall be of a minimum thickness of 0.05 inches, and either riveted or welded at their edges."

8 The design of the roof is also subject to the general paragraphs on stresses and details of the postal car specification.

9 There are several bills in Congress having in view the substitution of steel passenger equipment on railroads for present wooden cars. Should any of these become law, specifications for construction will be necessary, and, as the postal car specification has been approved and adopted as standard by the Government, no doubt this specification will be used as a basis in determining the requirements for other steel passenger equipment cars, not only for the roofs, but for the other features of construction.

No. 1388 *e*

SUSPENSION OF STEEL CARS

By E. W. Summers,[1] Pittsburgh, Pa.

Non-Member

If we could operate steel cars over rails having no kinks, curves or irregularities in their alignment, in other words, over an absolutely straight track, there would be little need of springs or other devices for flexible support.

2 Unfortunately the roadways we have to contend with cannot be made or maintained in true alignment. Frost and water make constant changes in the track support. Lateral curvature requires super-elevation of the outer rail. In passing from a tangent to a curve, or vice versa, the tracks under one truck are in wind with those under the other one, sometimes as much as 4 or 5 in. depending upon the degree of curvature and the length of the car.

3 Steel car bodies of the enclosed type, such as box cars, mail, baggage, or passenger coaches, are of rigid construction and have high torsional resistance. A three-legged stool on an irregular floor surface will stand upon all of its legs while one having four legs may carry all of its load upon two diagonal supports.

4 The use of truck springs helps the illusion that we are distributing the car body load on all of the wheels. The uneven deflection of the springs indicates directly the increased load of one spring over the other. When the track surface is warped more than the total spring travel, the whole load is carried at two diagonal corners, tending to twist the car body. This twisting tendency is constantly changing, first in one direction and then in the other, as the super-elevated rail changes from one side of the track to the other. The effect upon wooden passenger cars is to work the joints loose and cause them to screech and grind like the spokes of a wooden wagon wheel in hot dry weather.

5 The side bearings of steel sleeping cars pop like sledge hammer blows when the car is taking or leaving a curve. The slight twist in the track surface throwing excessive load upon two diagonal corners

[1]President, Summers Steel Car Company.

Presented at the New York Meeting, April 1913, of The American Society of Mechanical Engineers.

of the car causes the bearings to grip and adhere to each other coincident with the slewing of the truck. When the twisting of the truck exceeds the play in the parts around the truck bolster the side bearings let loose and jump with resulting hammer blows. More efficient roller side bearings may prevent the gripping and jumping, but the uneven load is still present. The twisting effect upon the car body is not removed.

6 Failure in roofs of wooden box cars and the resulting damage to merchandise in transit is due to this constant twist. Roof designers have attempted to remedy this by making the roof flexible and with slip joints. To be consistent they should go further and make the whole car of india rubber. A practical construction for the enclosed type of steel car bodies must and always will be rigid and of high torsional resistance.

7 The necessity for flexibility between the car body and the trucks, and for an even distribution of the load upon all of the wheels seems not to be fully appreciated as yet, but with each succeeding year wrecks due to broken rails, wheels and truck structure will drive this home. Suspension of steel cars, as has been developed by the writer in the past three years, does permit of a more even distribution of the load upon the wheels than with center-bearing trucks.

8 Fig. 1 is an illustration of a cross-section through an engine tender at the center of one of the trucks. It illustrates the method of suspension referred to and is applicable to any kind of car.

9 The inclined hangers *a*, the cradle *b*, and the side rockers *c* are shown heavily shaded. There are two inclined hangers at each side of each truck. A heavy rectangular bar extends through the lower ends of the hangers *a*. A cast-steel bracket, which is part of the car underframe, rests upon each end of the rectangular bar. The upper ends of inclined hangers *a* are supported upon the outer end of the cradle which rests upon the segmental rocker *c* and transmits the car body load directly into the truck side frame. The lower ends of hangers *a* are maintained a fixed distance apart transversely of the car, by reason of the brackets *d* being a fixed part of the car underframe. Their upper ends are held at a fixed transverse distance by their connection with the cradle *b*. Both the upper and lower ends of bars *a* are pivotally connected with rolling contact.

10 With one end of a car on level track and the other end having one rail at a higher elevation, the tendency will be for the high rail to carry all of the load at that end of the car, or to have the car support taken at two of its diagonal corners.

11 With *inclined hanger suspension* the car will swing sideways, the hangers at the high rail swinging inward and downward, while the ones at the other end of the cradle swing outward and upward, picking up the load at the low rail and maintaining its distribution on all of the wheels much the same as if suspended by two bars from

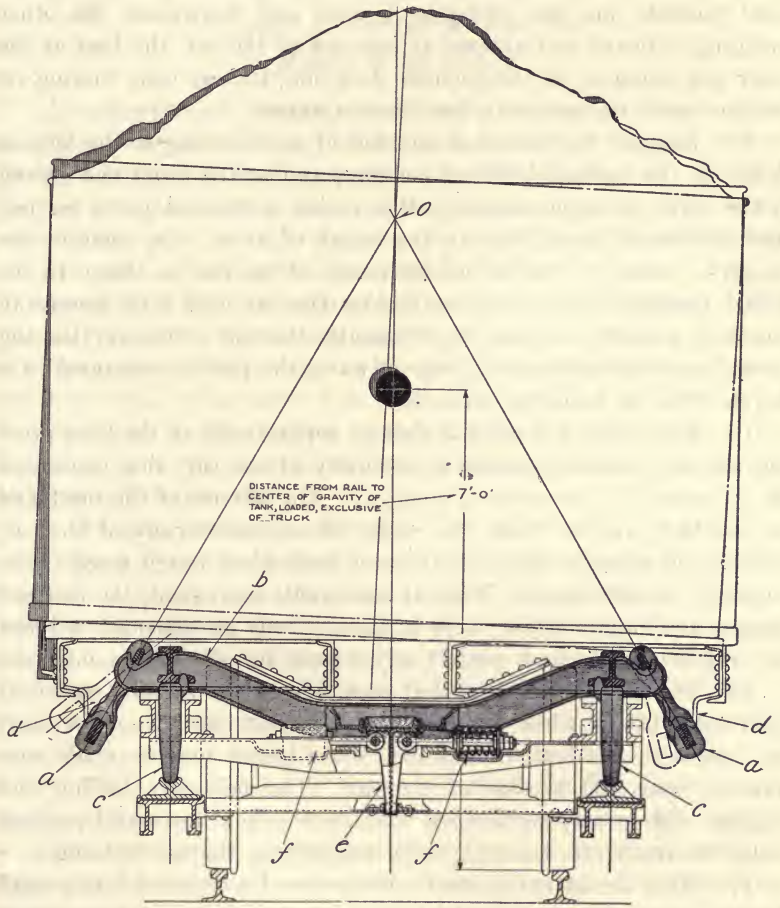

FIG. 1 CROSS-SECTION OF ENGINE TENDER AT CENTER OF ONE OF TRUCKS

a point *O* at the intersection of the center line of the inclined hangers extended. The load or rigid car body will find its own position vertically under this common point of support *O*, each of the extended suspension bars taking its share of the load.

12 The slight warping of the track surface, which causes all of the load to be carried on half of the wheels at two diagonal corners of a rigid car body with the ordinary center-bearing truck, is corrected by the short inclined suspension bars *a*, practically the same as if the suspension was from point *O*.

13 It is the inclination of these bars that makes vertical adjustment possible, one bar swinging inward and downward, the other swinging outward and upward at one end of the car, the bars at the other end swinging in the opposite direction, the car body finding its position much the same as a boat does in water.

14 Imagine the bars *a* at one end of a car swung to the left, as shown in the unshaded dotted position, and at the other end swung to the right an equal amount: this makes correction for a warped track surface of about 8 in. in the length of a car. Or, imagine the hangers *a* swung to the left at both ends of the car, as shown in the dotted position: this is the inclination the car body will assume in rounding a sharp curve at high velocity, the top of the car leaning inward and the bottom swinging outward, the position assumed by a bicycle rider in rounding a curve.

15 The cradle *b* is pivoted about a vertical axis on the king pin *e* and can also have movement transversely of the car, this movement being limited by the action of springs *f*. On account of the inertia of the car body and its load, the cradle moves transversely of the car, rotating the hangers about their lower ends when rough track is encountered at high speed. Without this cradle movement, the inclined hangers are impracticable; with it, the car body movement is without jar or jerk and we have perfect adjustment for all track conditions.

16 The car body is carried at each side almost directly under its rigid side girders, which by position have great depth and can carry the load with the least deflection. Floor beams may be made continuous from side to side of the car. The necessary buffing and tugging column may be disposed with its web in a horizontal position under the transverse beams, greatly simplifying the car framing.

17 With the advent of steel construction for enclosed cars a rigid structure came into use, one that cannot be handled over rough track as we have been handling the spongy wooden structure. There may be much hewing and chopping into old methods before the necessary compromise is made between the rigid car body and the changeable track surface, but why not do it all at once, and stop fooling with dynamite?

No. 1388 *f*

SIX-WHEEL TRUCKS FOR PASSENGER CARS

By John A. Pilcher, Roanoke, Va.

Member of the Society

Consideration of the subject of trucks for steel passenger cars is practically a consideration of trucks for any passenger car, the primary thought being that steel passenger cars should have steel trucks to prevent the possibility of fire, and also because of their great weight, metal is the most suitable material for strength and durability that can be used in the limited space available for the truck. The fire damage from a wooden frame truck could not be serious on a steel car, and there are wooden cars equally as heavy as the general run of steel cars; the writer having one in mind in the construction of which the sills were plated with 8-in. channels, weighing 172,700 lb. A few steel cars weigh as much as this, but we have no record of any weighing more. However, for steel passenger cars we will consider only the all-steel truck.

2 Practice of the past brings to our attention the pedestal type of passenger truck construction both for four-wheel and six-wheel trucks, the general characteristics of both being identical. The six-wheel truck with the same size axle is, of course, capable of greater load and also of transmitting to the car the track irregularities to a less extent, because the results of the irregularities are modified by the system of equalization. In the six-wheel truck the location of the equalizer springs is fixed at a definite point between the wheels.

3 While the details of these two trucks differ slightly, their functions are practically identical. Both trucks have been used for a considerable length of time, but the four-wheel truck was evidently developed first and its necessary functions, determined by experience, were later incorporated in the design of the six-wheel truck, which was probably first brought about by the increased loads.

4 Except for the especially constructed truck used by the Pennsylvania Railroad and one other, which we understand has been designed, these are the only regular types of trucks available.

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5 *Wheels.* For passenger service, the wheels have been practically narrowed down to steel tired wheels and wrought-steel wheels. The steel tired wheels have been of many forms of centers and fastenings; the latest recommended practice of the Master Car Builders' Association is that the tire be shrunk on and bolted. The recent development of the solid wrought-steel wheel has made available for passenger car service a wheel equally as safe and durable as the steel tired wheel at a very much reduced cost. The Master Car Builders' recommendations recognize both the 36-in. and 38-in. size in this wheel for passenger service, the 36-in., however, being the most generally used. These wheels, if carefully turned, should give as satisfactory service as any wheels available.

6 *Axles.* The standards of the Master Car Builders' Association gives the choice of selection of four sizes of axles:

Size of Journal	Axle Load, Lb.
3¾ in. x 7 in.	15,000
4¼ " x 8 "	22,000
5 " x 9 "	31,000
5½ " x 10 "	38,000

7 They also offer an axle as recommended practice with 6 in. by 11 in. journals for 50,000-lb. axle load. These loads, however, are for freight service; for passenger service we would recommend the use of from 60 per cent to 75 per cent of the loads used in freight service, based on the light weight of the car, and limiting the load to about 90 per cent of that in freight service, considering the weight of both car and lading. The lighter rating is, of course, to be taken for cars such as baggage and express, since the increased weight on account of lading would be heavier, while the higher rating could be taken for coaches and similar cars where the increase of the lading would be light. Table 1 gives the sizes of axles, and relative light weights of cars on this basis.

8 The Postoffice Department has limited the maximum load per wheel for postal cars to 15,000 lb. when using 5½ in. by 9 in. journals, and to 18,000 lb. when using 5 in. by 10 in. journals, making a further limitation based upon 18,000 lb. as the maximum brake load for any one cast-iron brake shoe under emergency conditions of brake application. This limitation of wheel loads, after deducting the weights of the wheels and axles, allows a pressure of 304 lb. per sq. in. projected area on the 5 in. by 9 in. journals, and 300 lb. per sq. in. projected area on the 5½ in. by 10 in. journals, also a pressure of 1522

lb. per lineal in. on the 5 in. by 9 in. journals, and 1665 lb. per lineal in. on the 5½ in. by 10 in. journals, and from the experience that some roads have had these seem to be just as high as should be allowed.

TABLE 1 SIZE OF AXLES AND WEIGHT OF LIGHT CARS

Axles, In.	Four-Wheel Trucks, Lb.	Six-Wheel Trucks, Lb.
3¾ x 7	40,000 to 52,000	60,000 to 78,000
4¼ x 8	52,000 to 72,000	78,000 to 108,000
5 x 9	72,000 to 100,000	108,000 to 150,000
5½ x 10	100,000 to 120,000	150,000 to 180,000

9 *Boxes and Contained Parts.* The Master Car Builders' Association has provided standard passenger boxes for axles with 3 ¾ in. by 7 in., 4 ¼ in. by 8 in., and 5 in by 9 in. journals. For the 5 ½ in. by 10 in. journal, which is often in use, they have not yet established recommended practices, but the previous designs are having their influence on the shape of the box for this journal.

10 *Pedestals.* Cast-iron pedestals seem to be usually the accepted material, and the Master Car Builders' Association has also provided standards to suit the boxes.

11 *Equalizer Springs.* These are four in number on both the four-wheel and six-wheel trucks, and while necessarily provided with a limited amount of deflection, they relieve the heavy truck frames of shock, and on six-wheel trucks provide the points of support for the proper equalization.

12 *Wheel Pieces or Side Frames with Transoms or Cross Ties.* These constitute the truck frame to hold the other parts in their relative position, and at the same time transfer the load from the bolster hangers to the equalizer springs. Being structures supported at four points, they necessarily have to be supported on springs to prevent excessive stresses due to any variations in the height of these four points. As an illustration, when the truck on a tangent is approaching a curve the rise of the outer rail is about 1 in. in 50 ft. This will raise one of these four points above the plane passed through the other three, and, while the difference is small in the short length of the truck, the irregularity has to be taken up by the springs, otherwise the truck frame would be similar to a four-legged table with one high leg.

13 When we consider the case of a derailment where one wheel of the truck, whether four-wheel or six-wheel, falls into a deep hole, or drops from a high rail, we find this condition exaggerated to such an extent that the whole load will be supported on two points. Then unless the structure is sufficiently flexible to follow, it will necessarily have to be strong enough to resist this abnormal load.

14 The calculation of the stresses under such uncertain conditions of loading is certainly a very complex problem. It is a pertinent question whether or not the designers should undertake to care for such an abnormal condition.

15 *Bolster Hangers.* The lateral movement of the bolster, one of the very necessary features of a passenger truck, is usually accomplished by the use of swinging hangers. This movement should be limited to from 1½ in. to 1¾ in. each side of the center, and in placing this limit arrangement should be made so that the stop will not be abrupt. This is ordinarily accomplished by the use of short hangers, or when long hangers are used by the addition of lateral motion springs, either of which offers an increasing resistance. Rollers on cylindrical or curved plains can produce the identical movement made by the short hangers.

16 *Bolster.* On the four-wheel truck the bolster is a simple beam, but on the six-wheel truck we have a more complex structure resting on four points of support. This condition brings up the same complex problem referred to in connection with the truck frame supported on four points, except that it rests on much more flexible springs than does the truck frame. These springs can hardly be expected to take up all of the variations in elevation that will likely be met with in case of a partial derailment. The same question as to whether or not the designer should allow for such abnormal conditions is again raised.

17 *Center Plate.* The usually accepted center plate for passenger cars is of the spherical pattern, allowing more perfect adjustment, and more even distribution of weights than can be obtained from the flat bottom center plate, but making necessary close and accurate adjustment of the side bearings to prevent the rocking movement between the car body bolster and the truck bolster.

18 The frictionless center plate would of course be very desirable, but conical rollers and balls of sufficient number, of the size that can be put in the available space, seem not to have been as successful as

could be wished. The ingenious designer is still at work on this particular problem.

19 *Side Bearings.* Side bearings must be made so that they can be readily kept adjusted to reduce to a minimum the rocking movement between the car body bolster and the truck bolster, and in this way confine the oscillation of the car to the variation in the deflection of the springs on either side.

20 The relative location of the side bearings, each side of the center, is a question often discussed. In passenger cars the practice generally is to place them at as great a distance from the center as practical. This in our judgment is correct, and of particular advantage in the case of frictionless or roller side bearings.

21 Where the side bearings are in actual contact and the bolsters are rigid, the oscillation of the car is controlled entirely by the difference in deflection of the springs on either side, so that if the side bearing is set out sufficiently far to prevent the car body upsetting on the truck, it serves its purpose in preventing car oscillation as well there as at any other location.

22 For the same type of side bearing, it offers just as much, but no more, resistance to turning than if located far from the center, because as the lever arm is increased the pressure is reduced in like proportion.

23 When the car on a tangent is approaching a curve, the rise of the track on the outer rail tends to bring a pressure on the side bearing of the leading truck, next the outside of the curve, and on the side bearing of the trailing truck toward the inside of the curve. Where the side bearings are in contact this variation in elevation has to be taken care of by the deflection of the springs which have to deflect the same amount whether the load is exerted on the bolster, at a point near the center, or far away from the center. If the load comes far from the center it takes much less pressure to influence the deflection of the springs. This would be to the decided advantage of the side bearings, particularly in the case of the frictionless side bearing, in preventing wear and would also, to a more limited extent, be of advantage to the ordinary flat side bearing.

24 *Brakes.* On passenger cars, the pressure on the brake shoes approximates the loads on the wheels. Particularly is this the case of coaches where the lading is only a small proportion of the total weight. In some braking arrangements the brake shoe load is even greater under certain conditions than the wheel load; therefore the lighter the wheel loads the better for the brakes. This is a decided

argument in favor of the six-wheel trucks for heavy cars, and an argument against the use of four-wheel trucks under heavy passenger cars, even though the weights can be readily sustained by the use of sufficiently large axles.

25 The application of the brakes to the six-wheel trucks in such a manner as to allow for the adjustment of worn shoes and worn wheels is a very difficult task on account of the limited space available. It is almost impossible to accomplish this task with the use of wheels less than 36 in. in diameter.

26 *Six-Wheel Trucks.* Since steel cars are of recent construction, and recent conditions have generally called for large cars, the weight is almost always great. The six-wheel, all-metal truck has the following advantages which make for its selection over other types:

a It is non-inflammable.

b It provides a strong material to resist the heavy loads, and occupies only a limited space.

c It provides a durable material.

d It reduces the axle loads, and the unit load on the bearings, lessening the liability to hot boxes, reducing the pressure on the brake shoes, lessening the tendency to heat the wheels and shoes, adding to the life of the brake shoes, and reducing the frequency between renewals and adjustments.

e It spreads the heavier loads over a greater area of structures, and brings more points of contact with the rail, reducing the influence of track irregularities on the riding of the car, and in cases of very heavy cars, where the unit pressure between wheel and rail might approximate the elastic limit, reduces the tendency to shell the wheel and roll out the rail, adding to the life of both.

27 It has been estimated that for a passenger car making 50,000 miles per year, the cost for hauling the car is 5 cents per lb. per year. If the six-wheel trucks weigh 14,000 lb. per car more than the four-wheel trucks necessary to carry the same car, it means the hauling of 14,000 lb. additional at a cost of $700 per year, which brings up a question for vital consideration.

28 While the wheels, brasses, and brake shoes, and other such removable parts may individually have a longer life, there are also more of them in service during the period. Careful comparison would have to be made to determine which has the advantage at this point.

29 *Four-wheel Trucks.* The four-wheel, all-metal truck is also available in connection with steel cars, and has the advantage of reduced first cost, reduced weight, smaller number of parts to maintain, and if the car is sufficiently light for the unit stress between the rail and wheel to be kept down to a point well below the elastic limit of the material, they should be given serious consideration. The only drawback under these conditions is the possibility of its reduced riding qualities. Its decided advantage in reducing the weight of the train should help to make it a favorite because of the corresponding reduction in the cost of transportation.

30 *Cast-Steel vs. Riveted Wrought-Steel Frames.* The introduction of heavy passenger equipment is rapidly doing away with both the four-wheel and six-wheel wooden frame trucks. The reduced cost of maintenance amply justifies this change if our information is correct. Cast-steel one-piece frames, and riveted wrought-steel frames of various cross-sections have been worked out and are now in use; both are reported as giving satisfactory service, but figures showing the exact relative cost of maintenance are not available.

31 The cast-steel one-piece frame has become a great favorite even in the face of the high unit cost of these particular castings. The adaptability of the castings to the various changes of form and section necessary on account of the limited available space has no doubt had much influence. The attractiveness of the one-piece structure, eliminating all joints, and furnishing a frame ready set up, is another strong argument in its favor. The manufacturers having control of this cast-steel truck frame have evidently been successful in reducing to a minimum the concealed flaws often met with in steel castings. This, no doubt, has added largely to its popularity.

32 While the absence of riveted joints and the consequent doubling of material at the joints, helps to keep down the weight, the fact that the working fiber stress of cast steel is taken low, and the sections at many points have to be made larger than is necessary on account of foundry limitation, the weight of the frame as a whole is great. This added to the large unit cost for special steel castings makes the user pay well for the advantages gained.

33 The riveted wrought-steel frame seems to have been held back in its development by the success of its rival in cast steel. Many users have shown conservatism in making use of the good thing already considered acceptable, hesitating to try out the different construction with the hope of lower first cost, with less weight, and equally good service.

34 Wrought steel at a very moderate unit cost has the advantage of being a very reliable material which can be worked to a relatively high fiber stress. The cost of fabrication, when the work is done in any large quantity, when added to the cost of material, will still leave a large margin in its favor. Is it possible that the lack of an especially interested advocate has prevented its virtues from becoming prominent, and delayed the experience needed to prove, in actual service, its worth?

35 We find that practically all of the prominent car builders have already worked up designs for wrought-steel trucks, and are ready to construct them if the purchaser so desires, but they do not seem inclined to push them, as they evidently offer no special inducement to their own advantage. Only a few have been built and placed under cars by them, and in some cases none, but from what I have been able to find out they have confidence in them.

36 I find several railroad companies building and using both four and six-wheel trucks, of the usual type of construction, with riveted wrought-steel frames, and from all reports they are giving satisfaction.

37 Another prominent railroad is using both four and six-wheel trucks, of a form of construction differing from the ordinary type, built of riveted wrought steel. As a large number of these are in daily evidence, and are constantly being built by them, they must be proving the worth of the riveted wrought-steel construction, as well as that of the special type of construction.

38 Experience of several years and careful comparison of the cost of maintenance will be needed to say whether the one-piece cast-steel frame, or the riveted wrought-steel frame truck will be the most advantageous, when both the first cost and weight are considered along with the cost of maintenance.

39 Variety of choice offers an opportunity for discussion. In the hope of bringing out this discussion we advocate for steel passenger cars: (*a*) Six-wheel truck; (*b*) the riveted wrought steel frame; (*c*) the use of the Master Car Builders standard axles, boxes and parts, and pedestals; (*d*) 36-in. wrought-steel wheels.

No. 1388 *g*

STEEL INTERIOR FINISH FOR STEEL PASSENGER CARS

By Felix Koch,[1] McKees Rocks, Pa.

Non-Member

Every one who has followed the progress in steel passenger car construction during the last ten years, which is about the age of the oldest steel passenger car, has noticed that very little, if any steel was used in the interior finish until within the last four or five years.

2 The first attempt to use steel in passenger cars resulted in steel underframes with wood superstructure. The next development provided steel underframe and steel superstructure, but with wooden roof and wood interior finish. Further developments eliminated the wooden roof and the final efforts produced an all steel car. Considering that this development was made during a period of four years, the results obtained are, to say the least, highly gratifying.

3 The earlier designs of steel cars with steel interior finish are sometimes called all steel cars, leaving the impression that they are fireproof in every respect, but this is not correct because too much wood was used in the form of wood furrings to enable the application of the steel finish with wood screws. These furrings were, of course, not exposed to view, but they nevertheless placed the cars outside of the classification "all steel cars." The idea that it was necessary to use wood furrings in order to make it possible to apply steel finish, or in other words, that wood screws had to be used, machine screws not being considered practicable, accounts to some extent for the tardiness in the introduction of steel in the interior finish.

4 The earlier specifications and designs for steel passenger cars made the use of machine screws for applying the interior finish prohibitive and impossible, which, of course, made it necessary to employ other means such as bolts or wood screws. Bolts for this purpose must have heads of special design to allow their insertion through slotted holes, etc., and to prevent them from falling through during the application of the nuts. The nuts, being exposed, are objectionable as they give

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an unsightly appearance, even if special cap nuts are used in place of the ordinary nuts, besides there are many places on a car at which it is impracticable to apply bolts. Therefore, to avoid machine screws and bolts the space between the outside sheets and the interior finish was filled with wood furring to allow for the use of wood screws. The objections to machine screws, caused by the belief that they would work loose in a short time, has, however, disappeared from experience gained through actual service as it has been shown that if set in white lead and properly applied they are entirely reliable.

5 There has always been and there still is a great difference of opinion as to how far it is advisable to substitute metal for wood in passenger car construction. The use of a small amount of wood in the interior finish, as for instance window sash moldings, seat arm rests, window capping, etc., should not be objectionable as it has certain advantages over steel which are desirable, but wood is used for such details to a considerable extent, and hundreds of cars are now in service in which the small amount of wood used in the interior finish cannot be detected except by an expert and such cars are to all intents and purposes fireproof cars, but the aim of many designers has been to eliminate the wood wherever possible on account of the many advantages possessed by steel, among which may be mentioned:

a Steel finish means non-combustion in case of fire.

b Steel prevents splintering in case of wreck.

c Steel finish can be easily removed should it become necessary to repaint the car at the inside surface of steel sheets, as the life of the steel car, to a certain extent, depends on the condition of the paint.

d Steel finish makes it possible to increase the interior width of the car where outside width is limited. This has been found particularly valuable in designing subway, elevated or suburban steel passenger equipment cars.

e Steel finish will avoid trouble which may be experienced due to different expansion of materials, steel against wood. This point need not be considered with steel and makes it unnecessary to provide for relief in all members of the finish running longitudinally, such as upper and lower deck sill moldings, etc. In fact, the steel finish has revolutionized to some degree the designs of wood finish in the wooden cars built since steel cars came in vogue. The cars of today are built on more sanitary lines, and

fancy moldings, fretwork and carvings have disappeared without losing sight of giving the cars an artistic finish, avoiding thereby lodging and breeding places for all kinds of germs which the world is fighting against today.

f Steel finish will, by comparison, be cheaper every year for the reason that it becomes more difficult to obtain the right kind of lumber for interior finish, which, of course, means increase in price of wooden cars.

g It is continuously becoming more difficult to obtain men who have had sufficient experience in applying wood interior finish, whereas it does not take the same experienced men for applying steel finish. A man requires from three to four years' apprenticeship to become an expert able to apply wood finish to a car, whereas an average intelligent man who is familiar with tools is able to become an expert in finishing cars with steel finish in from six to twelve months, and this fact of labor will have to be taken into account sooner or later.

h A more uniform color can be maintained on steel finish than on wood which comes in different shades, and it is very difficult and expensive to match perfectly all parts in one car with regard to shade without additional expense of glazing. Furthermore, the average life of paint applied to steel finish will be much greater than to wood finish for the reason that wood darkens with age. This, of course, influences the paint which is a disadvantage from the standpoint of illumination. Should it become necessary to repaint a car of wood finish, reworking of the finish by removal of the varnish and scraping is necessary, whereas in the steel finish the scraping is eliminated and the removing of varnish is alone required to be able to repaint the car.

i Steel finish is of advantage from a building standpoint in the handling and working up of material to make ready for application. Steel details can be worked up to a large extent before they are applied to the cars, which make it possible to manufacture the interior finish in much less time by the use of more men than it is possible to employ when applying a wood finish, as only a limited number of men have room to work at the same time in a car when the greater part of the fitting and cutting, etc., has to be done.

This has facilitated the establishment of a number of manufacturing concerns who devote their efforts almost exclusively to producing steel interior finishes not only for passenger cars but also for buildings. In addition to these any manufacturing company equipped with the necessary machinery for the making of drawn moldings, breaker presses, and ordinary welding and spot welding machines, is able to handle this class of work for railroads or carbuilders, who may not have the necessary equipment to do the work in their own shops and prefer to buy the interior finish as they buy other specialties.

6 All of these advantages are almost exclusively confined to the use of steel or other metals, although a composite material of a wood pulp nature or similar material made fireproof and waterproof by different processes, if applied in a proper way and used for ceilings and below the window sills, is not objectionable, and it may be applied in practically the same manner as steel.

7 The advantages possessed by wood over metal as a non-conductor can be very much reduced by the use of proper insulating material correctly applied. The use of proper insulation is of course of great importance and manufacturers of that class of material as well as railroads and car builders are giving a great deal of attention to the subject, and the time does not seem to be far distant when steel cars with interior finish of wood will be as scarce as steel passenger cars were ten years ago.

8 A great deal more could be said on this subject, but it is hoped that what has been brought out will show that steel interior finish has certain advantages not possessed by other material commonly used in passenger cars and that the disadvantages are few and not insurmountable.

No. 1388 *h*

PAINTING OF STEEL PASSENGER CARS

By C. D. Young, Altoona, Pa.

Member of the Society

A fundamental reason for painting any surface of a passenger car is to protect it from the damaging effects of the air which is more or less loaded with gases and moisture. For example, oxygen is destructive of iron and steel and when sulphurous gases are present they are quickly oxidized into sulphuric acid which is very corrosive to unprotected metallic surfaces. It, therefore, becomes necessary to protect the surface by a covering, and paint forms a substantial and convenient means for accomplishing this. If properly made and applied, it is an impervious coating, affording the needed protection by forming a hard waterproof, rubber-like sheeting or film which has sufficient elasticity to conform itself to the contraction and expansion of the surfaces to which it is applied. In addition to protection the surfaces may be beautified and embellished by the proper selection of pigments so as to bring about the harmonizing and artistic effects desired.

WOODEN EQUIPMENT

2 The painting of wooden passenger-car equipment has been, in the main, successfully accomplished, the painting schedule for the outside is briefly as follows: Apply two coats of primer, putty and glaze, followed by three or four coats of surfacers, as found necessary, after which the surfaces are rubbed down smooth with emery and oil, when two coats of shade color are put on. The necessary striping and lettering follows, completing by three coats of finishing varnish, consuming in all about sixteen to eighteen days.

3 The finishing of the interior of wooden cars generally has been in the natural wood, consequently it is only necessary to prepare the surface for the varnishing. A representative schedule which is used is as follows: One coat of filler, in paste form, which is sandpapered down to a smooth finish. Add one coat of rubbing varnish and rub down with sandpaper, after which apply three coats of rubbing varnish,

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and complete the finish, cutting down the gloss by rubbing with pumice and oil to produce the most pleasing "flat finish."

4 This method of finishing the wooden surfaces of cars has been attained with good results, so that naturally when the change to steel passenger equipment came some six years ago, a desire to retain as much past practice as possible seemed desirable. It was realized, however, that the all important point in the painting of iron or steel surfaces was to have the surfaces first thoroughly cleaned and entirely rid of scale and rust, as this is as necessary as the painting itself. To accomplish this, sand-blasting, where possible, was resorted to, supplemented by the use of wire brushes and emery cloth in the more obscure places and the more uneven surfaces. The sand-blasting, however, was confined largely to the outside surfaces and the latter practices to the inside portion of the car.

5 Iron and steel, while not presenting to the eye the same porous condition as wood, is full of finely divided pores, and the same atmospheric influences which enter the pores of wood and cause it to decay are ever ready to attack the unpainted surfaces of iron and steel, in fact the metal surfaces more readily combine with the oxygen and moisture of the air, forming what is rust or oxide of iron. Therefore, immediately after the sand-blasting and cleaning of the surfaces should come the application of the first or primary coat, as this is the most important one, from the preservative standpoint.

6 In the selection of a suitable primer it seemed but natural for the painter to be guided by the experience gained in the painting of locomotive tenders, and to follow the initial coats with practically the same process as with wooden cars, and I believe that so far as the subsequent coats are concerned, this practice was generally carried out by the earlier painting of steel passenger equipment. It is thought that an error has been made in this general practice, as will be explained later.

STEEL EQUIPMENT

7 The schedule for painting steel passenger car trucks, underframes and superstructures is as follows:

8 *Trucks.* Before assembling, all surfaces on truck parts throughout, including all concealed surfaces, but not including wheels and axles, must be covered with one coat of suitable primer. After assembling, all surfaces (except wheels) exposed to view after the body of the car has been placed on trucks, must be covered with two coats of truck enamel.

9 *Underframes.* During the process of construction, all parts of the underframe, including concealed surfaces and surfaces where metal bears on metal, must be covered with two coats of good metal preservative of a non-inflammable nature. All accessible surfaces must be covered with a third coat of metal preservative.

10 *Superstructures.* Before assembling, all parts made of iron or steel, including the roof, must be covered with one coat of primer. A second coat of primer properly thinned with turpentine, or similar material, must be applied to all surfaces, including those which are concealed when the car is completed. Wherever possible, this second coat must be put on after the sheets are in place.

11 After assembling, the outside of side and end sheeting, including letter plate and deck plate, must be covered with one coat of surfacer, the rough and uneven places glazed with "surfacer composition," four coats of surfacer being added, rubbed down with linseed oil and emery cloth, two coats of desired color material added, followed by striping and lettering, then finished with three coats of finishing varnish. The outside of the roof must be finished with one coat of heavy protective paint, followed by one coat of a mixture composed by volume of three parts of mixed ground color and one part of the protective coating used. The top surface and edges of headlining should be painted with two coats of some preservative, or color paint.

12 The interior of cars should receive very careful attention in order to produce the desired finish. To illustrate fully the various steps and time taken to complete the painting, the following is given as outlining the progress of the work. This is attained with the use of surfacers, colors and varnishes containing a relatively large amount of artificial driers and varnish gums, in order to obtain the artistic finish desired for the interior.

HEADLINING

1st day	Apply one coat and stipple after application.
2d day	Stand for drying.
3d day	Apply one coat and stipple after application.
4th day	Stand for drying.
5th day	Apply one coat and stipple after application.

SIDES ABOVE WINDOW SILLS AND ENDS

1st day	Apply one coat or priming.
2d day	Stand for drying.
3d day	Apply one coat surfacer.
4th day	Necessary puttying and glazing.

5th day	Apply as many coats surfacer as are necessary to make a level surface.
6th day	Same as 5th day.
7th day	Rub down with emery and linseed oil.
8th day	Apply one coat of ground color.
9th day	Apply one coat of ground color.
10th day	Apply one coat of ground color.
11th day	Apply one coat and stipple after application.
12th day	Apply one coat rubbing varnish.
13th day	Stand for drying.
14th day	Apply one coat rubbing varnish.
15th day	Stand for drying.
16th day	Apply one coat rubbing varnish.
17th day	Stand for drying.
18th day	Rub with oil and pulverized pumice stone.

SIDES BELOW WINDOWS

1st day	Apply one coat or priming.
2d day	Stand for drying.
3d day	Apply one coat surfacer.
4th day	Necessary puttying and glazing.
5th day	Same as 6th day.
6th day	Apply as many coats surfacer as are necessary to make a level surface.
7th day	Rub down with emery and linseed oil.
8th day	Stand, awaiting bringing up other work.
9th day	Stand, awaiting bringing up other work.
10th day	Apply one coat bronze green.
11th day	Apply one coat bronze green.
12th day	Apply one coat of rubbing varnish.
13th day	Stand for drying.
14th day	Apply one coat of rubbing varnish.
15th day	Stand for drying.
16th day	Apply one coat of rubbing varnish.
17th day	Stand for drying.
18th day	Rub with oil and pulverized pumice stone.

13 Formulae and panels for the various shade should be furnished the painters for their guidance in obtaining the shade of any of the colors which are desired.

RESULTS OF AIR DRYING PAINTS ON STEEL

14 The artificial driers and gums used in hastening the time of drying and hardening of the various coats and permitting the necessary rubbing continue this action so that the paints and varnish increase in hardness and brittleness, rendering them susceptible to

cracking and chipping, and the process of disintegration is aggravated by excessive expansion and contraction of the steel surfaces as compared with wood. The linear expansion of steel being more than twice that of wood would seem to indicate the use of more elastic coatings than formerly used for wooden cars.

15 This fact has been borne out in the service of the paint in a great many cases in an investigation which recently came under my observation. It was noticed that when some of the equipment had been in service about four months, the interiors of the cars were showing varnish cracks and checks. As time went on more cars gave evidence of this deterioration, the final outcome being that an investigation was made to see how serious the condition was. Some 400 cars were carefully examined, special attention being given to the selection of cars built by various manufacturers, where different makes of surfacers and varnishes were employed. An endeavor was also made to determine whether the cracking of the painted surfaces was confined to the varnish coats or the surfacer coats, or both.

16 In order to classify the various conditions found, four readings of percentages were arbitrarily taken, the condition of a new car being taken at 100 per cent:

Per cent	Condition of Varnish and Surface
90 to 80	Good, no checking
80 to 70	Fair, slight checking
70 to 60	Medium, considerable checking
60 to 50	Poor, checked from outside varnish coat to metal

Sample cars were selected to illustrate these various classes, and photographs were taken of the different defective surfaces so as clearly to indicate to the eye what the different percentages meant.

17 The result of this examination showed that the exteriors, including the sides, ends and vestibules, were in fair condition. There were a few exceptions to this, but they amounted to less than 6 per cent of the total having serious varnish and surface cracks. Interiors were found generally to be in a poor condition. About 80 per cent of the equipment examined had the varnish checked through to the surfacer.

18 Some of these conditions developed after four to eight months' service, indicating either that an entirely new system of painting would be necessary to overcome these troubles, or that a more elastic paint would have to be used for interior finishing under the present existing practice of painting steel.

19 To obtain some data indicating what should be done to meet the conditions, preliminary experiments were made by painting a number of panels and baking them in a heated oven. Repeated experiments along this line indicated that artificial driers could almost, if not entirely, be eliminated in the paint formulae and that more elastic materials could be used without the aid of artificial oxidizing agents. It was also observed that the elastic varnish used on the exterior of the cars could, under this system, be used to advantage on the interior, and by the aid of the heat of the oven they could be dried to the desired hardness, permitting the rubbing with oil and pumice to get the "flat finish."

20 The outcome of the experiments indicated that it would be desirable to extend the experimental panels to a full size car and, therefore, a proper baking oven was planned that would accommodate one of the largest existing steel passenger cars for the purpose of baking each coat as applied to the exterior and interior surfaces.

21 This oven, as designed and built by the Pennsylvania Railroad Company at its Altoona shops, is 90 ft. 3 in. long, 13 ft. wide and 15 ft. high. The frame work of this structure is made up of 3-in. I-beams for the sides, spaced 5 ft. centers. The roof framing is made of the same sections and curved to conform closely to the contour of the car roof. Each end of the oven has two large doors which can be readily opened and closed for the baking operation. The oven is lined on the inside with $\frac{1}{8}$-in. steel plate, and on the outside with galvanized iron of 0.022 gage. The 3-in. space is filled with magnesia lagging, thus effecting the needed insulation. The doors are insulated in a similar manner. Along the walls of the interior of the oven are placed 16 rows of $1\frac{1}{2}$-in. steam pipes, and along the floor, close to the walls, are arranged manifold castings with small lengths of pipe tapped into them at right angles. By this means over 2000 sq. ft. of heating surface is provided. A steam pressure of approximately 100 lb. to the square inch is used, thus making it possible to get an oven temperature of over 250 deg. fahr. Rectangular openings, made adjustable, are provided on the sides near the floor line, allowing the necessary admission of air for circulation. Four 8-in. Globe ventilators are spaced at equal distances in the roof, likewise provided with dampers to regulate the size of the opening. By this means of ventilation, fresh air, which is required for the proper drying of paint, is obtained. as well as providing for the egress of the volatile matter present. Automatic ventilation and steam regulation have not, at the present

time, been applied, but these have been considered advisable, if the result of the experiment seems to warrant a more extended application of the practice.

22 A track is placed on the floor of the oven and connected at each end of the oven with other tracks leading into the regular paint shop where the different coats of paint are aplied to the car before each baking operation.

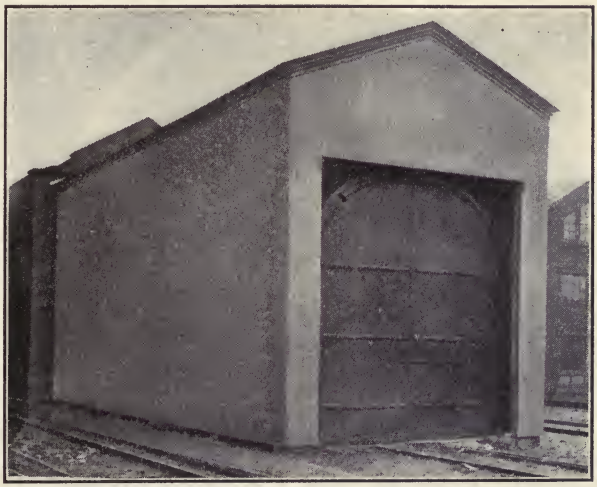

FIG. 1 EXTERIOR APPEARANCE OF OVEN

23 Photographs of the general appearance of this oven from the outside, and one end of the interior with a car within the oven are shown in Figs. 1 and 2. Fig. 3 shows the steam piping in detail.

BAKING PAINT ON STEEL

24 The outline of painting a car in this oven is briefly as follows: First, a priming coat is given the exterior and interior of car, which is then moved into the oven and baked for three hours. The temperature at the start is about 160 deg., but rapidly rises at about 1 deg. per min. until a temperature of 250 deg. is reached, requiring about 1½ to 2 hours. The oven is held at this temperature until the lapse of 3 hours, when the car is withdrawn, allowed to cool sufficiently to work upon, after which the surfaces are glazed and depressions and

uneven places puttied. The car then receives its first coat of surfacer, is returned to the oven for 3 hours, baked and removed for additional coats which vary from two to three in number as the needs of the case require.

25 After the last coat of surfacer has been applied and baked, the outside surface of the body of the car is rubbed down with emery

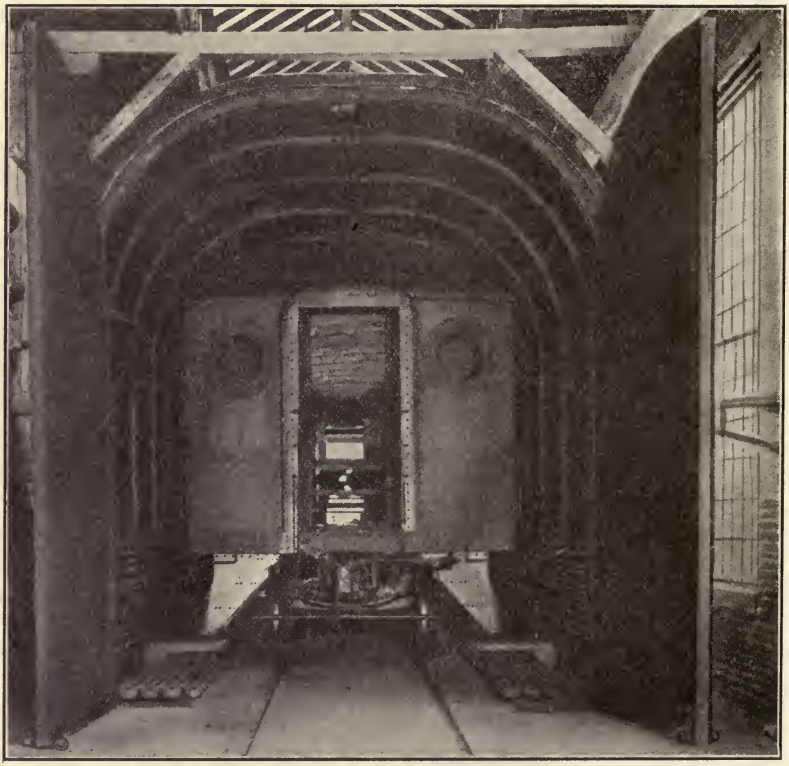

FIG. 2 VIEW OF INTERIOR OF OVEN SHOWING CAR IN PLACE

and oil to produce a flat and smooth surface. The various color coats used, such as tuscan red on the outside, pale green, bronze, and bronze green on the inside, are then put on. Two coats of each color are required to get standard shades. Each coat of color is likewise baked.

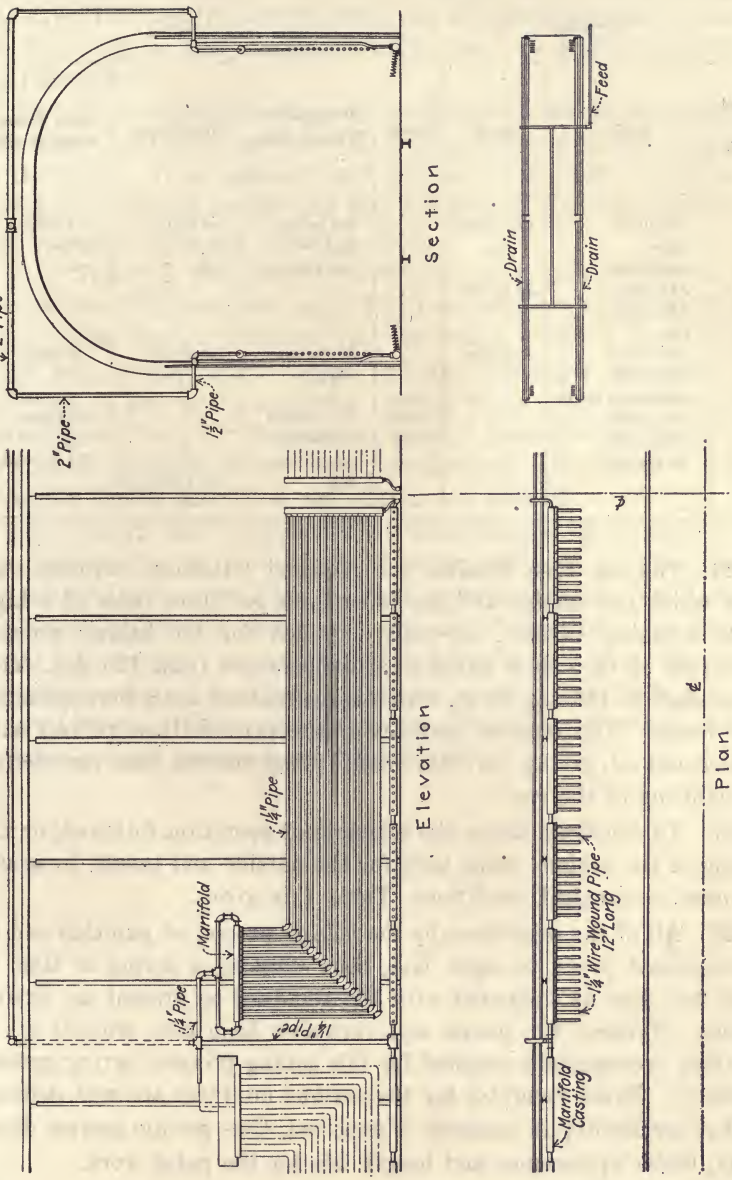

FIG. 3 DETAIL OF STEAM PIPING

TABLE 1 TIME SCHEDULE FOR PAINTING EXTERIOR AND INTERIOR OF STEEL PASSENGER CARS

	Outside			Inside		
Period of Work	Body	Roof	Trucks	Body Above Window Sills	Headlining	Body Below Window Sills
1	1st prime	1st prime	....	1st prime	1st prime	1st prime
2	glaze	....	....	glaze	glaze	glaze
3	1st surface	....	....	rub-ground	rub	rub
4	2d surface	2d prime	....	.....	.....	.....
5	3d surface	....	....	.....	.....	.....
6	rub	....	....	.....	.....	.....
7	1st tuscan	3d prime	....	2d ground	1st green	1st green
8	2d tuscan	....	....	stipple	.....	.....
9	stripe and letter	....	....	.....	.....	.....
10	1st varnish	....	truck	1st varnish	.....	2d green
11	2d varnish	....	color	2d varnish	.....	.....
12	3d varnish	....	....	3d varnish	.....	1st varnish
13	.....	....	....	rub	2d green air dry	.....

26 The car then receives the required lettering, striping, etc., after which the outside and inside surfaces get three coats of a high grade finishing varnish, especially adapted for the baking process. Each coat of varnish is baked at a temperature from 120 deg. fahr. at the start to 150 deg. fahr., which is maintained until the expiration of 3 hours. The interior surfaces of the car are then rubbed with pumice and oil, giving the "flat finish" effect desired, thus completing the painting of the car.

27 To illustrate better the schedule of operation followed, or the timing of the various coats, both for the outside and inside, to secure the most economical conditions, Table 1 is given.

28 All of the work done by the baking process of painting can be accomplished in six to eight days, thus effecting a saving in time of about ten days as compared with the standard or present air drying system. Further, the paints and varnishes have been worked up so that they are especially adapted for this baking process, having greater elasticity. Exact formulae for the various mixtures are well defined, so that uniformity in material is expected, thus giving greater durability, better appearance and longer life for the paint work.

29 The checks and cracking previously found will be considerably lessened, if not almost removed. By oven painting the work is done

under more uniform conditions, which at the present time are so hard to control. It enables the surfaces of the car to be heated uniformly and dried thoroughly, thus removing any objectionable moisture before the first priming coat is applied, which is a very desirable feature of the new method.

30 A considerable saving will be effected by the shorter time that cars will be held out of service when undergoing repairs and repainting in the shops. It is expected that dirt, soot, etc., will not adhere or imbed themselves so readily and that the general appearance of the car will be improved by the baking method.

31 This oven was placed in service the early part of this year and the results of the complete car at this time seem to justify the experiment. They seem to indicate that the results obtained from a small panel can be duplicated in the full size passenger equipment car and that, if this is the case, this method of painting can be used to advantage not only for the painting of steel passenger equipment cars, but for the painting of any other full size steel structure of a similar character where protection and finish are desired.

32 Results and indications at this time seem to justify our expectations that the new process of baking will give, over the present air drying system: (*a*) Longer life of material applied; (*b*) a general appearance as good or better; (*c*) less cost of material at no increase in the labor charge; (*d*) complete sanitation for old cars; (*e*) a considerable saving of time for shopping cars, which results in a saving of shop space. These advantages are offset by the initial cost of installation and operating cost of the oven.

No. 1388 *i*

PROVISIONS FOR ELECTRIC LIGHTING IN STEEL PASSENGER CARS

By H. A. Currie,[1] New York

Non-Member

Hardly more than perfunctory attention is, as a rule, given to the lighting equipment of a car by the car designer. After all other apparatus and equipment are taken care of, the lighting is considered and fitted as well as possible into the remaining space.

2 From a standpoint of practical consideration of the welfare of passengers, the lighting plays one of the most important parts; therefore, every effort should be made to arrange the light units so that no discomfort be occasioned, and to install the apparatus and wiring so that operating failures be reduced to a minimum. In this connection I might say that the United States postal authorities at Washington are going into this subject very carefully at the present time to insure fair treatment for their postal clerks in the railway mail service; very stringent requirements have been ordered both as regards general illumination and reliable performance.

3 The two essential considerations for the designing engineer to keep in mind in laying out his installation are: (*a*) The arrangement of parts in a manner to allow of easy inspection and repair; (*b*) protection against mechanical injury. Convenience and accessibility of apparatus, fixtures, junction boxes and wiring mean much to the inspector. It is a well-known fact that the average inspector will pay little attention to those parts which are difficult of access, and much better inspection work will result where parts are arranged in an accessible manner. It is of equal importance that the various parts be protected in such a manner as to avoid all possibility of injury to them while the car is in service. The other essential features of the lighting installation are discussed in the following paragraphs:

4 *Axle Generator.* The usual practice is to suspend the generator by swinging links at the inside end of the truck, and belt it to a pulley

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on the axle. For mounting the axle pulley a straight machined seat should be provided in all cases if electric lighting is planned or can be anticipated. Until recent years the universal practice was to provide the regulation tapered axle and allow the manufacturer of electric lighting equipments to adapt his pulley to an unsuitable seat in the best manner he could. The belt and pulley troubles which resulted were disproportionate to any possible advantage from retaining the tapered axles. It is customary for the manufacturer of lighting equipments to provide his own supporting structure adapted as circumstances permit for attachment to the truck. The resulting suspension is at best something of a makeshift.

5 It would be a consummation much to be desired if truck designers would provide a generator support built integral with the truck; the requirements are not difficult and it is certain that the generator builders would be glad to make their machines conform to the truck builder's suspension. As the matter is now handled, nothing causes them more delay and inconvenience than obtaining the numerous details of truck and underframe construction necessary for making an intelligent layout of the generator suspension. In designing the suspension it is desirable that the space required for the belt be kept as clear as possible. The end tie of the truck frame, if used, should not be deep and should be located at a level that will make it possible for the belt to straddle it. Outside brake beams when used are a necessary evil from the standpoint of generator location. Head room for the generator should be considered in laying out deep center girders, brake rigging and piping. All the open space that can be provided about the generator is desirable because it facilitates thorough inspection. The generator terminal board should be attached to the underframe of the car close to the generator and readily accessible.

6 *Battery Box.* On account of the obvious necessity for convenience in handling the heavy batteries, the battery box location has practically been standardized. As the weight and dimensions of elements are almost identical, it is unnecessary to change the hanger design after a satisfactory arrangement has once been used.

7 *Charging Receptacles.* The charging receptacles have been allotted a permanent location on electric lighted equipment. Care should be taken to arrange the wire leading to the receptacles to prevent interference with brake rods, etc.

8 *Switchboard and Regulator Lockers.* (*a*) The switchboard locker should be so located as to be at all times easily accessible to the

trainmen; no pains should be spared in the design and installation of the board; nothing but fireproof material should be used. A receptacle for spare lamps and a report card holder are convenient accessories. (*b*) The regulator locker is generally located under the switchboard and on the generator end of the car. Good ventilation is a necessity. Provision against dampness and dirt is imperative. The regulator lockers should be fitted with locks to guard against accidental or wilful interference with apparatus. In designing lockers for lighting apparatus it is recommended that liberal space be provided so that changing of equipment, repairing, inspecting and testing can be done to the best advantage.

9 *Conduit.* In steel-car construction, metal conduits are almost universally used. In the better type of steel car the interior conduits can be concealed behind metal molding and suitable outlet boxes designed to harmonize with the contour of the molding. Some designers are satisfied to have exposed conduit used exclusively throughout the car. In laying out wiring conduit, direct runs without sharp bends should be used. Care in locating the conduits will facilitate the installation of wires and prevent damage from moisture, etc.

10 *Fixtures.* Where side lighting is used, a satisfactory arrangement can be obtained by designing the fixture to meet the contour of the molding. In center deck lighting, it is advisable wherever possible to arrange the carlines so that a direct support to each fixture may be obtained. On platforms provision for one or two-lamp outlets is sufficient. A plain socket mounted on the platform ceiling has been used in some instances. A better arrangement would be a metallic reflector sunk flush in the ceiling.

11 *Emergency Lights.* It was formerly customary in applying electric light to retain gas lighting as a reserve. Increasing reliability of electric lighting apparatus has made this unnecessary and in the best present practice no gas equipment is provided. For emergencies it is customary to provide holders for candle lamps; but it is only on rare occasions that these have to be used, if the electric equipment is of a good modern type.

No. 1388 *j*

PROVISION FOR ELECTRICAL EQUIPMENT ON STEEL MOTOR CARS

By F. W. Butt,[1] New York

Non-Member

In providing for the electrical equipment on steel motor cars, several important points should be considered. On account of its metallic construction, the car becomes a negative conductor, or, in other words, the car is grounded, and all electrical apparatus must be well insulated against leakage of the electrical current.

2 Switches, circuit breakers, fuses, etc., should be so located that the arc when opening a circuit will not reach the metal structure of the car. In cases where space is limited, and it becomes necessary to locate circuit breaking apparatus in such a way that there is danger of the arc reaching the metal structure, suitable arc shields of non-conducting and non-inflammable material should be used.

3 Switches, terminals and other apparatus, having exposed live parts, should be protected against accidental contact by enclosing them in boxes or cabinets. This protection is most important where apparatus, such as mentioned above, is located in or near the space which is occupied by passengers.

4 It is sometimes found necessary on account of the restricted space in toilet rooms, motormen's cabs, postal and baggage compartments, etc., to attach electric heaters directly to the sheathing; the heater coils then are necessarily close to the sheathing, and as a means of protection to the paint and varnish thereon, an insulated backing should be applied between the sheathing and the heater.

5 Particular attention should be given to locking bolts, nuts, screws, etc., to prevent them working loose on account of vibration, especially those which are used to secure the apparatus. The vibrations of the motor gearing are transmitted to all parts of the car and they are more pronounced when the motor suspension lug is mounted on the truck transom, without the use of suspension springs. Vibration

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is more easily transmitted through the solid structure of steel cars than it is in cars of wood.

6 In the design of new cars it is sometimes found convenient to locate various members of the structure, especially in the underframe, so the apparatus can be suspended from them without the use of intermediate supports. This is desirable as it is often found that many parts can be omitted from the car. Where heavy apparatus is to be suspended from intermediate supports, large heavy members are required, sometimes complicated in design in order to obtain clearance between parts of the structure or apparatus.

7 Where it is possible, apparatus hangers should rest on the members which support them and not depend entirely upon a vertically bolted or riveted connection. The hangers should be well braced, especially those which hang far below the underframe, to prevent swaying of the apparatus, due to the motion of the car. The hangers can be so designed as to provide the necessary bracing, but to accomplish this odd shapes are often required which increase the cost of manufacture. It is then desirable to provide hangers and separate braces of simple design.

8 When several switches, fuses and other electrical apparatus are required for the motor, control and auxiliary circuits, large switchboard area is necessary, and in some instances, the switchboard has been installed in one of the end bulkheads, occupying most of the space between the body corner and door posts. In recent steel cars, intermediate body end posts are used as part of the general scheme for anti-telescoping provisions at the end of the car. These posts extend from the body-end sill to the body-end plate, and it is recommended, in order to interfere as little as possible with the general anti-telescoping scheme, that two small switchboards be used, one placed in the bulkhead on each side of the body-end door opening, and located as high above the platform as the size of the boards will permit. This arrangement of switchboards provides for the use of short intermediate body-end posts, extending upwards from the body-end sill to the horizontal frame member, located just below each switchboard and connected to the body corner and door posts.

9 In wooden car construction it is necessary to provide ground wires from the various electrical circuits to some part of the car which is a negative conductor. This is unnecessary on cars of steel construction, as the electrical circuits can be grounded at almost any part of the car structure.

10 The steel car is safer than cars of wood construction, as there is no danger of bad fires on account of short circuits. Parts of the structure of a steel car will not become alive, as is sometimes found in cars of wood construction.

11 The wiring conduit on a steel car should be provided for at the time the car is being designed. Unless this is done, difficult bends in the conduit may occur and it is sometimes found necessary to cut and reinforce the structural members.

No. 1388 *k*

AIR BRAKES FOR HEAVY STEEL PASSENGER CARS

By A. L. Humphrey,[1] Wilmerding, Pa.

Non-Member

Advancement in the development of air brakes has been no less contingent upon the development of rolling stock than the economic handling of traffic through the use of heavier and faster trains is contingent upon the advancements in motive power. A review of the history of railroad transportation development in this country will show a steady and unceasing advance from year to year. Equivalent advancement in the efficiency of appliances such as air brakes was consequently necessary in order that the control and safe handling of longer and heavier trains should not operate as a barrier to these developments.

2 A brief comparison of the conditions existing at the time of the introduction of the air brake with the conditions at present, will show that the advancement in rolling stock has been more rapid than those who have not been in close touch with the situation are likely to realize. For example, the weight on drivers of high-speed passenger engines has increased from 25,000 to 180,000 lb. The drawbar pull of locomotives has increased from 7000 lb. to 30,000 lb.; working steam pressure has increased from 125 lb. to 225 lb.; weights of passenger cars have increased from 20,000 lb. to 150,000 lb. The schedule speeds of passenger trains have increased from 30 miles per hour to 65 miles per hour, and it is not uncommon for speeds to reach as high as 85 to 90 miles per hour.

3 Taking the average weights of trains and average speed at the time the air brake was introduced as compared with the trains and speeds of today, the weight per vehicle has not only increased nearly eight times, but the foot-pounds of energy to be destroyed is nearly 15 times as much. In order to meet the demands of modern service conditions as efficiently as heretofore, means should be provided for dissipating the total energy stored up in this swiftly moving mass in

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at least as short a time and distance as before. In fact it is desirable to do this in as much less time as is consistent with comfort to passengers and accuracy of control, in the case of service stops, and in as much shorter distance or time as may be possible in the case of emergency. Not only must the brake be automatic in its operation, but it must be capable at any time and under any conceivable circumstances to produce the maximum possible retarding force within as short a period of time as the known resources available and physical limitations will permit.

4 When we consider that it requires a distance of 8 to 12 miles for a locomotive of modern design, hauling a train of say ten cars, to accelerate to a speed of 80 miles per hour and that this same train should be brought to a standstill within the shortest possible time—or say in one-tenth of the distance required to accelerate to this speed—it is hardly conceivable that this can be done with the means available, which is a retarding force produced by frictional contact of metal shoes against the wheels, which is in turn limited by the adhesion between the wheels and the rail.

5 This factor, viz., the friction obtainable between wheel and rail and shoe and wheel is the basis on which we must start, and upon which we are limited, as to the amount of retarding force obtainable. It is therefore of first importance in designing an air-brake installation to give due consideration to the contact between the wheel and rail and the possible efficiency of the brake shoe. The air brake in itself is practically limitless in the amount of force obtainable, but the practical application of this force is where the line must be drawn. In this connection it is worthy of note that the brake shoe today has about four times as much work to do as it had 30 years ago. The chief effect of this, however, is to destroy the brake shoe at a much more rapid rate, without permitting any material lengthening of stopping distance.

6 The improvements made in air brakes in recent years, which have made it possible to control the present heavy high-speed passenger trains with approximately the same degree of efficiency as the older forms controlled the equipment of their day, have been based on scientific principles and experience in obtaining reliable information and data. The matter of time of transmission of compressed air was not so important a factor with the shorter trains and slower speeds as it is today, where a train running at 80 miles per hour passes over a distance of 117 ft. per second; consequently a few seconds saving in the time of getting the brakes to fully apply is just so much relative gain

in the time and length of stop. With the latest improved pneumatic equipment, the maximum brake cylinder pressure can be obtained throughout a modern train of ten cars in 4 seconds, which is the shortest possible time that this can be obtained by serial quick action through a train of this length. For the purpose of shortening this time serious consideration is being given by some railroad officials to the type of brake equipment used on the New York subway, and known as the "electro-pneumatic," which would not only tend to cut the time of full application in two, but by means of the electric control all brakes are applied simultaneously, which not only assists in shortening the stop but in preventing shocks, etc.

7 Another equally important factor now coming more prominently into use is the application of brake shoes to each side of the wheel, known as clasp brakes. The virtue of clasp brakes, however, is not so much in the aid they afford in shortening the stop as in the equalizing effects of pressure on the wheels, journal box bearings and trucks, the minimizing of lost motion which affects the brakes through increased piston travel, and the less tendency toward wheel sliding while the brakes are applied.

8 While a comparison of the relative merits of a brake equipment, as with most mechanical devices, is frequently based on their maximum capacity, it must be borne in mind that an air-brake equipment must be designed to include flexibility for service operation, in which it is operated 99 per cent of the time and during which time it should be capable of handling smoothly the extreme lengths of trains, while at the same time it must be capable and ready under all conceivable circumstances to produce the maximum permissible braking force in case of an emergency.

9 It is not especially difficult to increase the speed of a train from 30 to 40 miles per hour, but it requires a vastly greater amount of energy to increase the speed from 60 to 70 miles per hour. In like manner, for any given increase in speed, the additional amount of work required of the brakes increases proportionally. If, therefore, the brakes for the heavier trains and higher speeds of today permit of stopping in about the same distance and with the same flexibility of control as could be done with brakes 40 years ago, and with the trains of that period, it is at least gratifying to know that the advancement made in this particular line of railroad development has kept pace as closely as it could consistently with the development in transportation facilities, through which its rate of advancement is largely controlled.

No. 1388 *l*

CAST-STEEL DOUBLE BODY BOLSTERS, PLATFORMS AND END FRAMES FOR STEEL CARS

By C. T. Westlake, St. Louis, Mo.

Member of the Society

Cast steel as applied to underframes and end frames of railroad cars is the result of careful design and painstaking, and thorough development of the art of casting in sand molds. These large steel castings are made in baked molds, confined in massive metal forms, by a special method that assures positively against swelling due to pressure of the inflowing metal, and yet permits yielding to the pressure of the contracting metal when cooling, so that the castings are very accurate in shape and close to size, and are free from shrinkage stresses.

2 Steel is an alloy of iron and carbon and differs from other alloys of iron by being capable of developing all its physical properties to the maximum degree. Its most distinctive properties are rigidity, ability to stand maximum forces without yielding; elasticity, ability to return to normal after being loaded to deflection; ductility, ability to stand distortion beyond its elastic limit without fracture; malleability, permitting it to be forged; tensility, high tensile strength; and weldability, permitting it to be welded by heating and hammering. These properties which steel possesses in a maximum degree distinguish it from all other alloys of iron.

3 Cast steel and rolled steel are produced by the same processes and of the same materials, are of the same chemical composition and have the same physical properties, and cast steel may be substituted for rolled steel, using the same fiber stresses, and its substitution is limited only by the minimum section that can be poured in the molds.

4 As recently as 1893, cast steel was comparatively unknown in car construction, and in that year its introduction began in the use of truck bolsters for freight cars. This was followed a few years later by body bolsters or transoms, and it was only after their use on

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freight cars had demonstrated satisfactorily the reliability of the material and design, that attention was turned to passenger cars.

5 The double body bolster was first to receive consideration for passenger cars, and although, due to casting difficulties, its weight was at first excessive, it was quickly refined and assigned to its proper place with other cast-steel articles. It was found to be so much lighter, stronger and permanantly effective than the built-up type, by forming a one-piece cradle or support for each end of the car body, that its use soon became almost universal in construction of passenger cars.

6 As the demand increased for stronger, safer and less combustible cars, the problem of replacing wood with steel developed many difficulties. The wooden car was the result of many years of experimenting, of cutting and trying with a material easily worked, but as one of the most valuable properties of cast steel is its adaptability to combining a multiplicity of complex parts into a single one of simple form, it was gradually developed from the double body bolster form, first to include end sills, then end and buffing sills; next the end and buffing sills were combined with longitudinal members extending to, and connecting with the double body bolster. Finally these parts, together with many others combined into a single simple member at each end of the car underframe, and comprising so many of the fixed parts that it is now only necessary for the car builder to connect them by center girders and to apply draft and buffing gears and the superstructure to complete the car frame.

7 The ideal underframe should have all connecting members in the same plane so as to avoid buckling due to eccentric leading; it should be so designed that each member will independently perform its individual functions, passing the stresses from one member to the other through the smallest number of properly aligned connections; and all should be so arranged in relation to each other as to form one powerful, compact, shock-absorbing element throughout the length of the car.

8 This can be accomplished to great advantage in cast-steel construction since the metal can be properly distributed in proportion to the stresses. The gusset plates can be placed in the same plane as the flanges of intersecting members, and the whole reduced to minimum weight and to the smallest number of parts with practically no joints. It can be molded to any desired conformation, can be shaped to any curve, useful or ornate, without the use of expensive dies, can be provided with necessary projections joined to the main members by

proper fillets. Openings may be provided with finished and reinforced edges, and all parts may be molded to symmetrical, pleasing contour, all edges rounded and a complete, practical, operative device, emanating from a single source furnished to the car builders ready for application.

9 As the rounding of curves necessitates the use of convex ends to the car body, the central portion of the ends is most exposed and liable to receive initial impacts, and this portion should be made strongest and most capable of properly transmitting the force of impacts to the balance of frame.

10 The underframe receives the force of end collision as a column load on its longitudinal members, while the end frame receives it as a transverse load on exposed members supported at their ends. As it is impracticable under these conditions to make the end frame equally as strong as the underframe, provision should be made for protecting the end frame against destructive forces. The underframe should be arranged so as to receive the initial impact, and if the encountered force is sufficient to destroy it, it should fail in such manner as to form additional protection to the end frame.

11 This is accomplished in cast-steel construction by arranging the parts of the longitudinal members so that when loaded to destruction by a collision force, the end portions yield upwardly, thus folding the exposed portion of the platform up against the end of the car body, and forming an addition to the end frame to assist in distributing the force to all the longitudinal members of the superstructure. The advantage of this construction has been demonstrated in wrecks when this identical action has taken place, the safety of passengers assured, and the property loss kept low.

12 The cast-steel platform as now provided for blind end cars, comprises the buffing sill having recesses for the buffer foot plates, holes and brackets for the buffer stems, pockets for the buffing device, brackets for safety chains, lugs for draft gear, brackets for drawbar carry irons, anti-telescoping plate, extensions of the center sills and bottom chords of the side sills, all of the double body bolster members including side bearing arches and extending for a distance of over 14 ft. inward from the end of the car to a point considerably back of the truck center, and counting rivets, gusset plates and connecting angles, combining more than 1000 pieces into a single, powerful, shock-absorbing element of less weight than fabricated material of the same strength.

13 The cast-steel platform and double body bolster for vestibule cars comprises all the parts enumerated for blind end cars, and in addition, includes the exposed platform longitudinal members, step risers and end sill, measures over 17 ft. in length, is made of a single piece, and is also of less weight than fabricated material of the same strength.

14 Since the government has taken a hand in the construction of cars used in its service, stronger body end frames are being used, and as the end of the car is the first to encounter end collision forces, it reasonably deserves closer and more careful consideration.

15 Most damage is produced by end collisions and to protect life and property from them, the colliding object must be prevented from entering the car. To accomplish this, the end frame and end portion of underframe should be constructed so as to distribute the force of collision into all the longitudinal members of the car, passing it into the largest mass, utilizing every particle of available inertia to absorb the force without permitting it to reach and act upon the contents or occupants of cars.

16 The end frame proper should be designed so that when a single member is loaded, all will act with it, and this can be accomplished only by connecting them so as to form a single mass, and best by forming them in a single piece as in cast-steel construction.

17 In designing the cast-steel end frame we assume it to be a beam supported at its upper and lower ends and loaded at a point about 18 in. above its lower end. We provide connections between the end frame and balance of car frame of sufficient value to develop the full transverse strength of the end frame; the vertical members of end frame are connected by horizontal members so that in case the end frame is loaded to destruction the connections are sufficient to disrupt all the longitudinal members of the car frame, and when they yield all parts will be forced toward the center of the end of the car and tend to prevent one car telescoping the other.

18 Cast-steel parts weigh less than built-up members carrying the same load since the metal in castings can be properly distributed in proportion to stresses. In built-up construction the metal overlaps at the joints and this, together with the rivet heads, makes an additional weight which in cast construction is avoided. In the latter, reliance is placed in a single solid member and, as there are no joints, there is no chance of their being imperfect or becoming loose.

19 The advantage in cast steel to the car builder is also very

great. To produce a platform of the built-up type at least eight different classes of material are required. This comes from eight different manufacturers, frequently from as many different points of production, much of it in less than car load lots, and all has to be requisitioned, purchased, received, stored and recorded for use on each particular lot, and in order to reduce storage space and avoid congestion in the car plant, all deliveries have to be carefully and accurately timed, and followed up. Then each material has to be passed through the different departments of the car plant to be cut, shaped, punched, drilled and the same timing and tracing methods used, so as to have all parts completed at the proper time. When cast steel is used but one material is purchased from a single plant, only one piece is handled, that in car load lots, and when it arrives it is immediately ready and available for application without storage or re-handling, facilitating completion of the car by leaving more car plant machinery available for other work.

20 A plant capable of producing castings of this nature in quantities to meet requirements of the many car plants must have buildings of extensive area and equipment in proportion, as it ordinarily requires about ten days for a casting to pass through the various processes of casting, cooling, roughing, cleaning and machining, and an accumulation of ten days' output has to be constantly accommodated. All handling and conveying apparatus must be in duplicate so as to insure uninterrupted operation and machines for finishing must be of the highest grade and maintained in perfect condition to produce accurate and proper results.

21 In car construction cast steel stands preëminent as the best material for reducing to the minimum the weight and number of parts while maintaining requisite strength and other essential properties, and its popularity and use will proportionately increase as its benefits and advantages become more generally recognized.

No. 1388 *m*

SPECIAL ENDS FOR STEEL PASSENGER CARS

By H. M. Estabrook, Dayton, O.

Member of the Society

After the passenger car had emerged from the stage-coach type of construction the box-like shape of car was introduced with straight longitudinal floor sills and with straight vertical side and end posts. These members were of wood, the ends of the longitudinal floor sills being tenoned into mortises in the wooden end sills. The vertical side and end posts were in like manner tenoned into the side and end sills at their lower end and likewise into the wooden side and end plates at their upper ends. These side and end posts were maintained in their several positions, by wooden spacing blocks or bridging, and the whole structure tied together by means of iron rods and bolts. These spacing blocks served further the double purpose of affording a foundation for securing the outside panels and the wooden interior finish.

2 Several types of roof were quite prevalent in early passenger car days, among them being the round top or omnibus roof, which has again made its appearance in steel passenger cars in some parts of our country. Another type of roof was the Ogee, or turtle-back, and later came the monitor, or raised deck roof. The prevailing type of hood projection over the platforms was the "duck's bill" type, as illustrated in Fig. 1, which also furnishes a good idea of the framing employed in those days. Fig. 2 shows end framing of these same cars.

3 A little later the projecting platform hood was changed from the "duck's bill" type to the bull-nose type. Figs. 3 and 4 show respectively a longitudinal section and exterior of these cars. Fig. 5 shows the end construction and Fig. 6 the standard framing employed in the first bull-nose hood cars in the early eighties. Up to the middle eighties no systematic attempt had been made to strengthen the ends of cars. The platform members were all of wood and the end framing of the car had not experienced much change in the way of strength-

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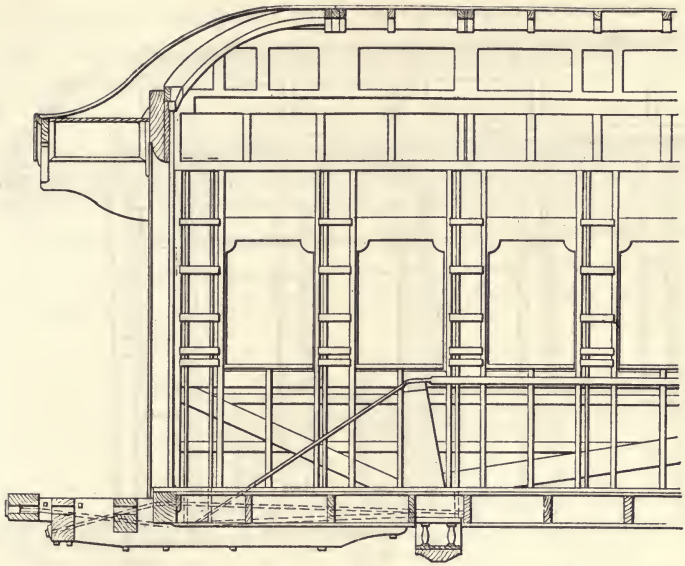

FIG. 1 "DUCK'S BILL" TYPE OF HOOD PROJECTION OVER PLATFORMS

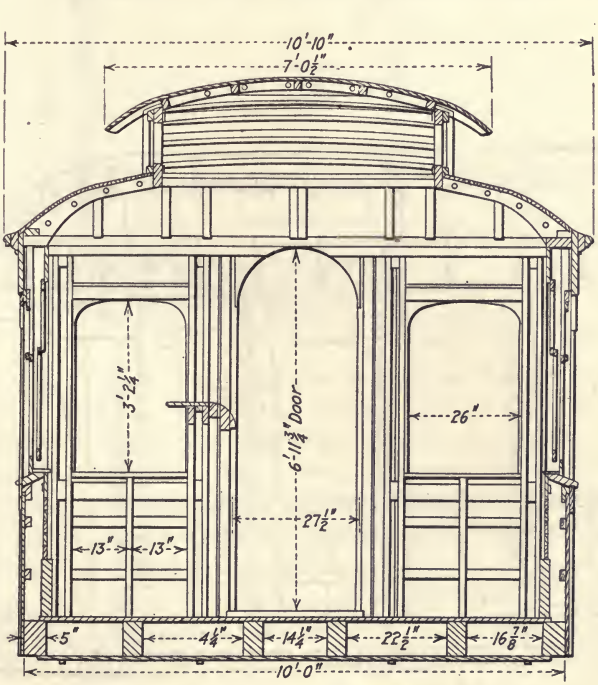

FIG. 2 END FRAMING FOR "DUCK'S BILL" HOOD PROJECTION

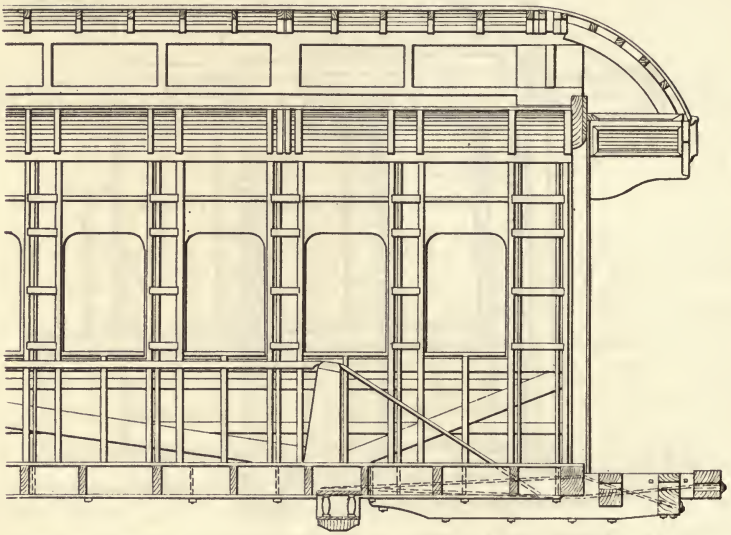

FIG. 3 SECTION OF BULL-NOSE TYPE OF CAR

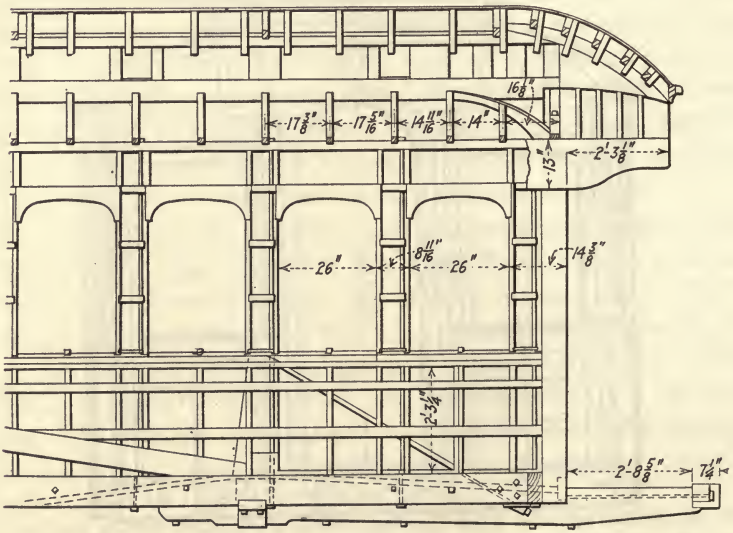

FIG. 4 BULL-NOSE TYPE OF CAR, FRAMING EXPOSED

ening from the earlier types. With the advent of the narrow vestibule in 1887, which was immediately followed by the broad vestibule in 1888, came the demand for a stronger end.

4 About the year 1890 there was brought into use what was known as an "anti-telescoping" end framing. This construction consisted of double side and end sills with a steel plate 8 in. by ½ in. from 18 to 24 ft. long, sandwiched into the double side sill, with the end of these plates turned so as to form a foot against the end sill. The double end sill had a steel plate 8 in. by ¾ in. and the length the width of the car, sandwiched into the end sill. The end posts of

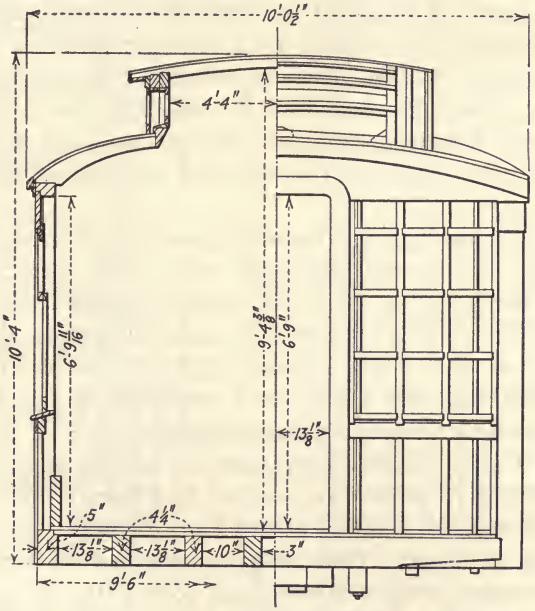

Fig. 5 End Construction of Bull-Nose Type of Car

the car were reinforced by steel bars 3½ in. by ¾ in., extending downward through and bolted through the sandwiched end sill and having their upper ends extending upward and bearing on and bolted through a steel plate 6 in. by ⅜ in., which was bolted to the oak end plate of the car. This stiffening plate extended across the width of the car and the ends of the steel plate being turned so as to form a foot upon the side plate of the structure. This anti-telescoping construction is illustrated in Fig. 7. This design of end framing came into general use throughout the country and is in use today in the

majority of wooden passenger cars built since 1890. It is interesting to note that this anti-telescoping framing is the same, with some modifications and additions, as was adopted by the United States Government for the construction of full postal cars and known as Specification No. 1.

5 Somewhat later this type of end framing was elaborated upon by the use of a heavy steel angle flitched into the end sill, with the end still further reinforced by a 20 in. by ½ in. steel gusset plate on the under side of the sills, and by the use of steel Z-bars in the end posts and a heavy steel angle introduced into the construction of the end plate of the car.

6 The increased weight of the vestibules and anti-telescoping end framing developed the necessity for a stronger platform construction than the old style wooden platform member that had been used for many years. About the year 1895 the standard steel platform, composed of steel I-beams, came into general use, and was employed continuously until the advent of the steel car superseded it by other designs.

7 Notwithstanding the frantic efforts of Congress toward the general adoption of steel passenger cars, it has been stated upon reliable authority that no vestibuled wooden passenger car, in the construction of which was employed the anti-telescoping end framing, in a straight-on end to end collision (although frequently having the ends concaved) has ever had the end crushed in to the extent of the adjoining car body telescoping and entering it.

8 The United States Government in seeking to strengthen the end construction of postal cars adopted this form of anti-telescoping end framing with the addition of two 7-in., 23.46-lb. steel bulb beams on either end of the car. These bulb beams have their flat base resting against the outside of the reinforced end posts of the car, being located in line with and immediately behind the vestibule diaphragms and face plate. At its lower end, this bulb beam has the head and web notched out with the base flange extending downward through the flitched end sill, the main body of the beam resting upon the 1 in. thick steel plate on top of the buffer beam. At the upper end these bulb beams have the web and bulb head sheared diagonally so the base flange extends upward on the outside of the end plate of the car framing, and through this flange passes the top piston stems of the vestibule mechanism. This type of construction is now obsolete in postal cars, Congress having enacted a law requiring them to be of steel construction.

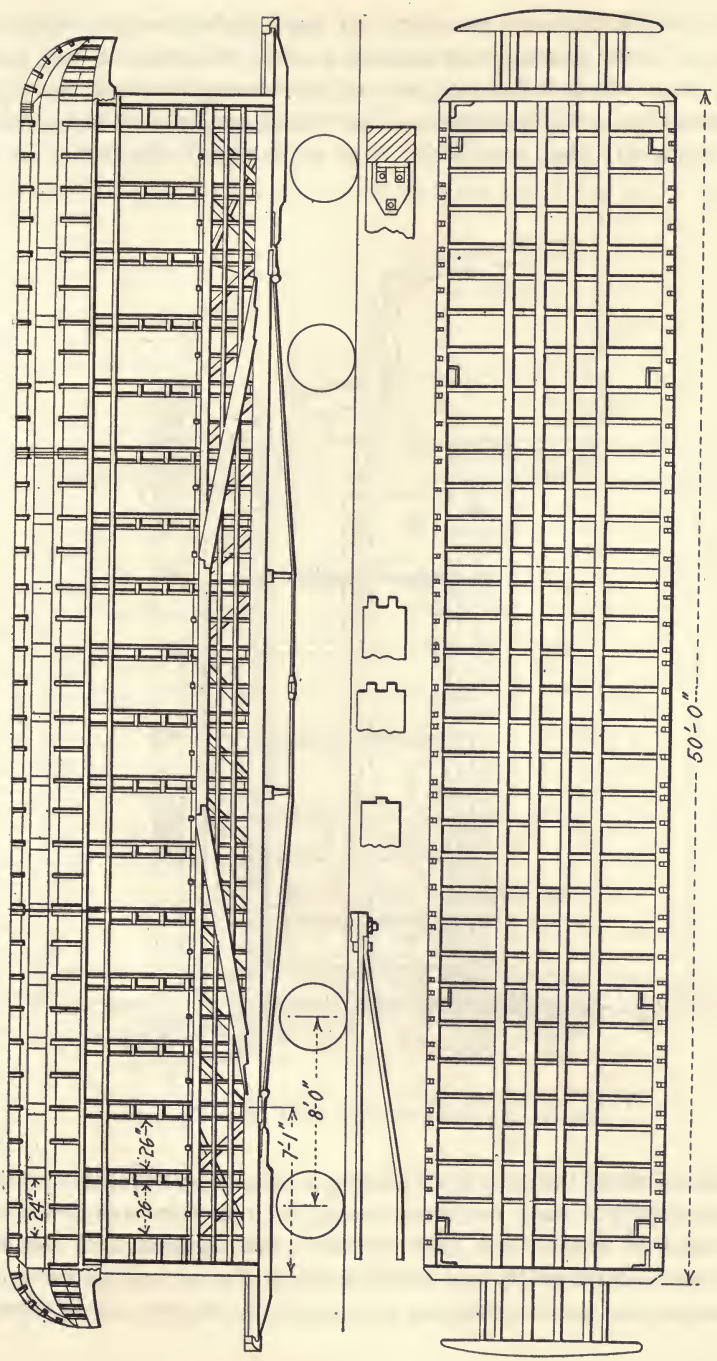

FIG. 6 STANDARD FRAMING EMPLOYED IN FIRST BULL-NOSE CARS

9 When the steel passenger car made its appearance about the year 1905, the passenger car entered a period of transition and evolution from which it has not yet entirely emerged with a recognized standard form of construction. The wooden car had attained a degree of uniformity that established it as an accepted standard. In the

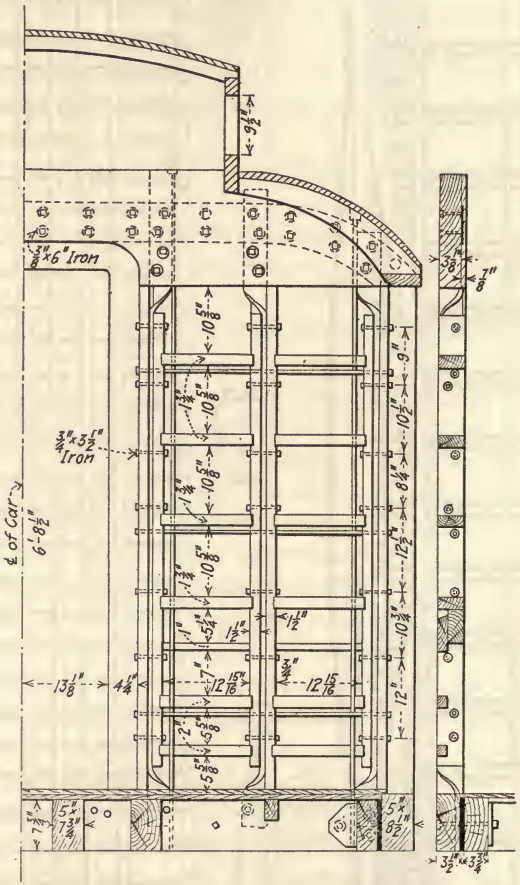

FIG. 7 ANTI-TELESCOPING IRON END FRAMING

construction of the early steel passenger cars, as was probably natural, an attempt was made to follow closely the lines employed in the construction of wooden cars, with the result that the first steel cars were inferior in strength of end construction to the prevailing wood construction, but the evolution has been rapid, one improvement following

close upon the heels of another. In the entire history of car building, there has probably not been devoted so much concentrated thought and study to the improvement in design, by the most expert engineering talent of the railroads and car builders, as has been shown since the introduction of steel cars. This has resulted in rapid improvement of end construction until we have today reached a design

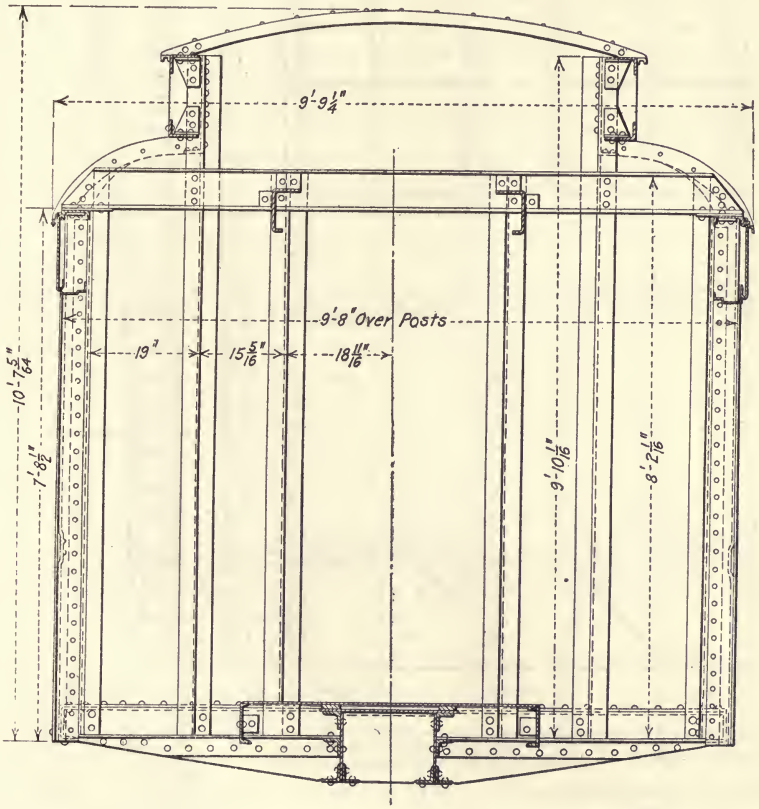

FIG. 8 BODY END FRAMING, TYPE SHOWN IN FIG. 9

that is considered practically standard. This development has no doubt been hastened by the action of Congress relative to steel postal cars and the coöperation of committees of the railway mail service, the railroads and the car builders, to the end that a standard specification for the strength of the various parts of the car, and especially

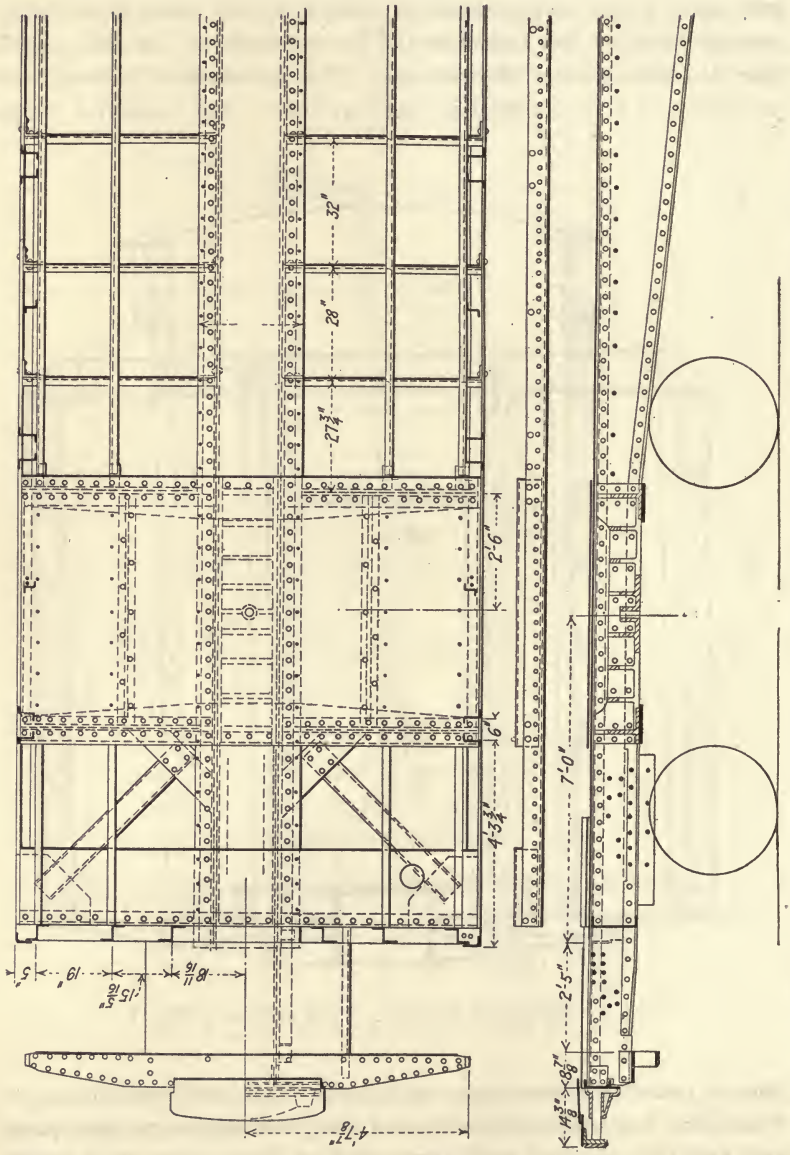

FIG. 9 STEEL CAR CONSTRUCTION, CENTER SILLS EXTENDING FULL LENGTH OF CAR

the end construction, has been adopted by the Postoffice Department in which it is provided that:

The maximum end shock due to buffing shall be assumed as a static load of 400,000 lb. applied horizontally at the resultant line of forces acting as the center line of the buffing mechanism and at the center line of draft gear, respectively, and shall be assumed to be resisted by all continuous longitudinal underframe members below the floor level, provided such members are sufficiently tied together to act in unison.

The sum of the section moduli of all vertical end members at each end shall be not less than 65 and the section moduli of the main members, either forming or adjacent to the door posts, shall be not less than 75 per cent. of this amount. The horizontal reactions of all vertical end members at top and bottom shall be calculated from an assumed external horizontal force, applied 18 in. above the floor line, to all vertical members in the proportions given, such force being of sufficient amount to cause bending of all vertical members acting together, and top and bottom connections of vertical members shall be designed for these reactions. Except where vertical end members shall bear directly against or be attached directly to longitudinal members at either top or bottom, the assumed reactions shall be considered as loads applied to whatever construction is used at end sill or end plate and both these last named members shall have section moduli, respectively, sufficient to prevent their failure horizontally before that of the vertical end members. All parts of the car framing shall be so proportioned that the sum of the maximum unit stresses to which any member is subject shall not exceed the following amounts in pounds per square inch—these stresses, unless otherwise stated below, are for steel having an ultimate tensile strength of from 55,000 to 65,000 lb. per sq. in.:

Bolsters of Rolled Steel—Stress shall not exceed 12,500 lb. per sq. in.

Sills and Framing of Rolled Steel—Stress shall not exceed 16,000 lb. per sq. in.

When cast steel is used the allowable stresses may be the same as for rolled steel except tension stresses, which must be at least 20 per cent less than those allowed for rolled steel, as specified above.

10 To meet these requirements, there are at this time three distinct forms of construction employed: The one most generally employed is illustrated in Figs. 8 and 9, which is composed of rolled-steel sections with the center sills running the full length of the car from buffer beam to buffer beam. Another type is that in which the rolled steel center sills connect at the bolster with a steel casting, forming a combined body bolster, center and side sills, and end sills, as illustrated in Figs. 10 and 11. Another type is that in which the rolled-steel center sills connect at the bolster with a steel casting, forming a combined body bolster, center and side sills, end sill and the entire end frame of the car, as illustrated in Fig. 12.

11 In the first form of construction, shown in Figs. 8 and 9, rolled sections are employed entirely. The members forming the

center sill construction extend the full length of the car from one buffer beam to the other and all other longitudinal members, such as side sills, belt rail, etc., extending the full length of the car body and in the case of vestibuled cars, the rolled section side plate extends the full length of the car from one vestibule corner post to another. The end sill is usually composed of pressed or rolled shapes riveted to the center-sill construction and extending laterally outwards to the sides of the car, the ends of the side-sill members butting against and being riveted to these end-sill members. The upper end plate of the car is composed of rolled or pressed sections extending in one piece across the width of the car and attached to the longitudinal side plates by connecting angles and gussets. To this end plate are also attached the longitudinal members of the upper deck sides. The end posts are rolled or pressed sections, usually Z-sections, extending downward to the bottom line of and riveted to the end sill. The upper ends of these posts extend upwards to the top line of and are riveted to the end plate. The nose piece or buffer beam is composed of rolled channels with their flanges turned inwardly towards each other, presenting their smooth surfaces on the outside, these channels being formed to suit the contour requirements of the vestibule, the channel members forming a box construction with top cover plates.

12 This buffer beam extends across and is riveted to the outward ends of the center-sill construction, from which it will be observed that the purpose of this design is to transmit the end buffing shock to the center-sill construction. The vestibule corner posts are rolled channel or Z-sections, whose bottom ends extend down into and are riveted to the outer ends of the buffer beams and whose upper ends are riveted to the vestibule end plate and to the upper longitudinal side plate of the car body, which projects beyond the end of the car body to meet and to connect with this vestibule corner post. The center vestibule posts are 6-in. I-beams whose lower ends extend downward through and are connected to the buffer beam member and whose upper ends extend upward to and are connected to the vestibule end plate steel angle. Between the upper ends of these center vestibule posts and the end of the car body, are longitudinal compression members in the form of steel channels or angles. These rolled section corner posts, door posts and vestibule door and corner posts, are encased in light steel casings formed to give them the finished appearance of the same members in a wooden car.

13 In stub-end cars of this type of construction, the buffer beam

is of considerably heavier construction than in the vestibule car, and is usually composed of a built-up box construction or a one-piece steel casting, this buffer beam being secured immediately to the outside face of the end sill. In this construction there is usually employed a

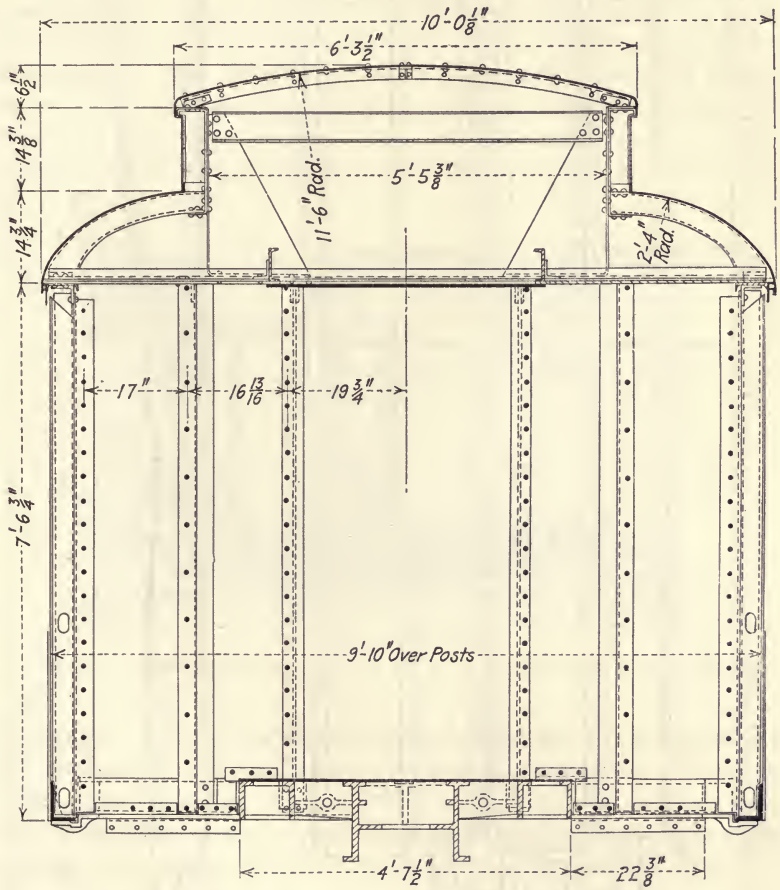

Fig. 10 Body End Framing, Type shown in Fig. 11

much heavier vestibule center post than in the vestibuled car. These vestibule posts, usually being a 12-in. I-beam, are located immediately in line with and behind the vestibule diaphragm and face plate. The end-post construction is much the same as described for the vestibuled car, there being a difference, however, in the construction of the end plate, which in the stub-end car is a pressed channel section formed

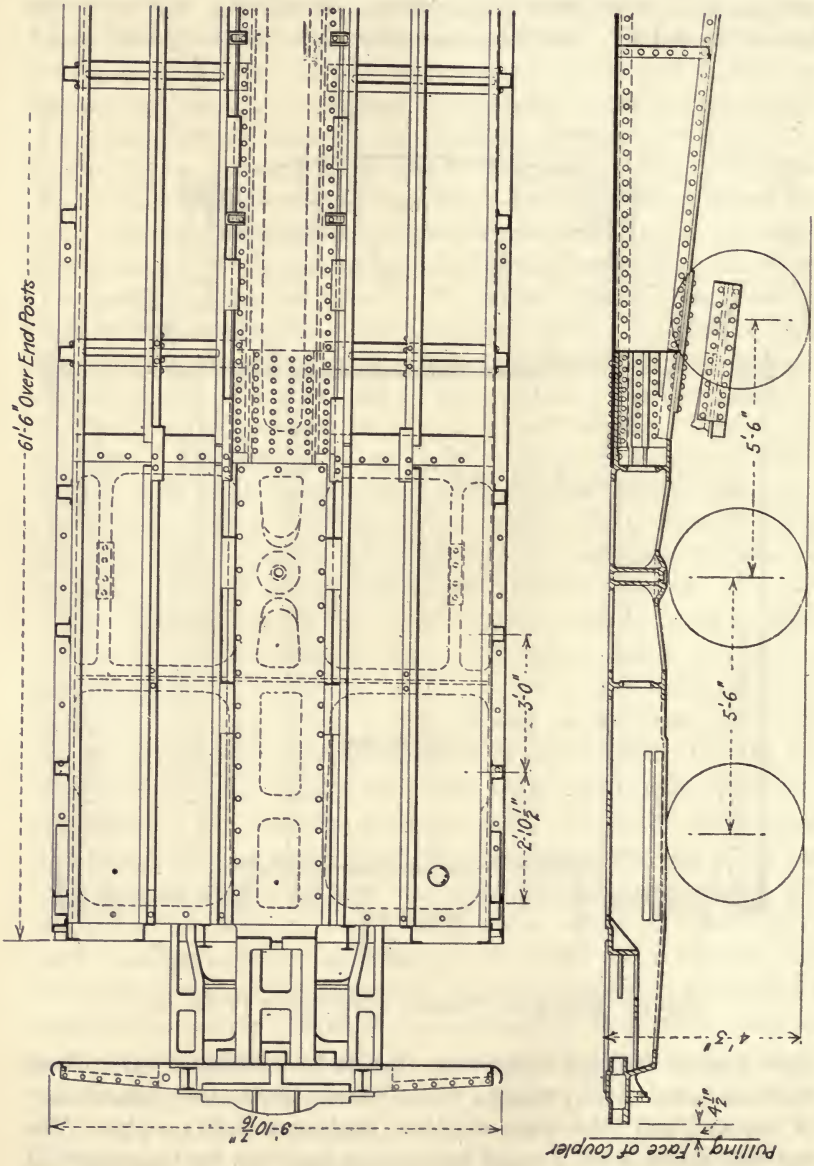

FIG. 11 STEEL CAR TYPE, CENTER SILLS CONNECT AT BOLSTER WITH A STEEL CASTING

to suit the contour of the car end, this channel end plate being placed across the end of the car in a horizontal plane, and into and riveted to this channel end plate are the upper ends of the corner posts, end posts and vestibule posts.

14 In the second type of construction referred to, a steel casting is employed forming the body bolster and platform to which the center-sill construction is riveted to this steel bolster. This construction is illustrated in Figs. 10 and 11, from which it will be observed that the center sill construction, the end sill, platform and buffer beam are all embodied in one steel casting. The end-post construction, the corner posts, vestibule corner and center posts are practically of the same

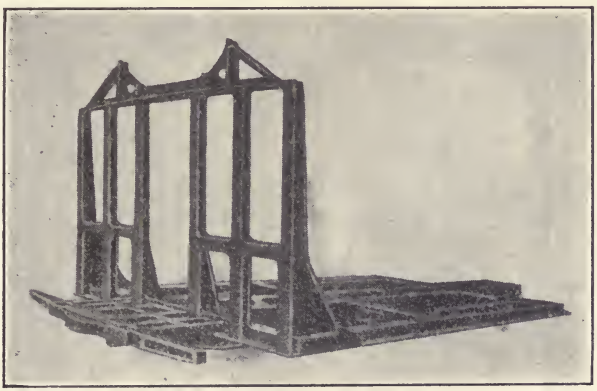

FIG. 12 INTEGRAL STEEL CASTING USED IN END FRAME CONSTRUCTION

construction as described for the built-up type, the difference being in the method of attaching the lower ends of these posts. The steel castings have openings or pockets in the end sill and buffer beam members, in which the lower ends of these posts rest and are riveted to the casting. The construction of the end of the car body, the vestibule and hood are substantially as described for the built-up construction.

15 This type of construction for the stub-end car is substantially the same as that just described, the exception being that the steel bolster and end-sill casting takes the place of the built-up type of center and end-sill construction, the end post, corner post and upper end construction being identical in the two types.

16 In the third type of construction referred to the entire bottom framework of the car from the bolster outward to the platform and

buffer beam, is one integral steel casting, and the entire end framing of the car is one integral steel casting, as illustrated by Fig. 12.

17 In referring to the three types of construction just outlined,

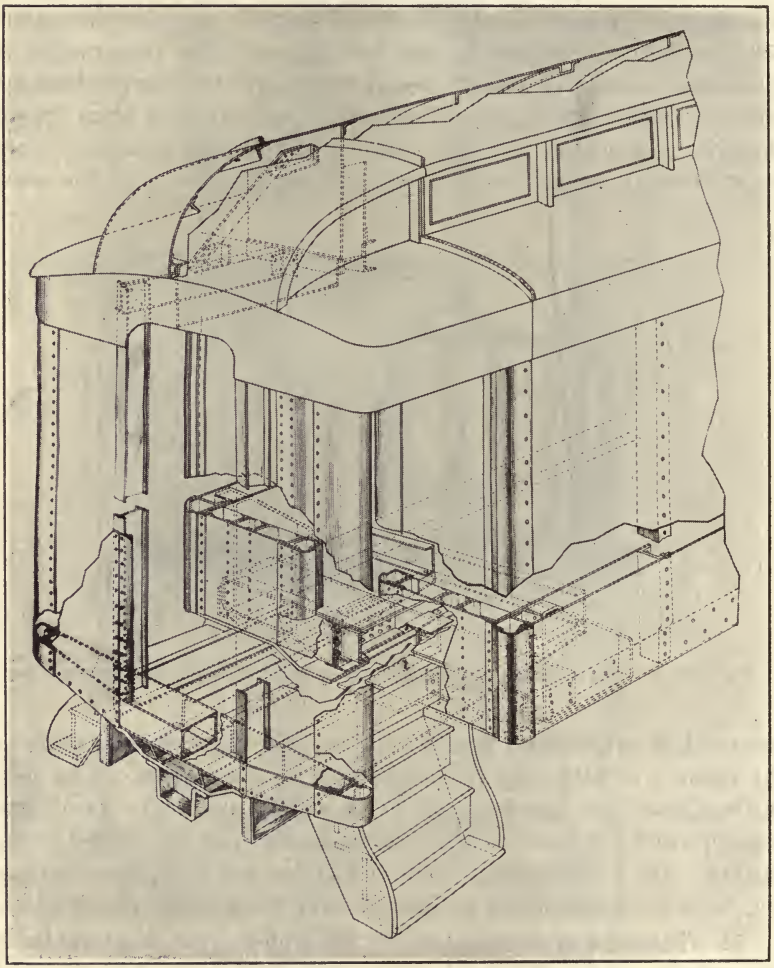

FIG. 13 COLLAPSIBLE VESTIBULE CONSTRUCTED ENTIRELY OF STEEL

it must be understood that reference is made to them only as types, and no attempt is made to describe the construction of any one railroad or car builder in particular, or to undertake to establish any of

the forms described as being a standard, as the details of construction vary to a considerable degree with different railroads and builders.

18 It is of course apparent that the weight of the steel car is

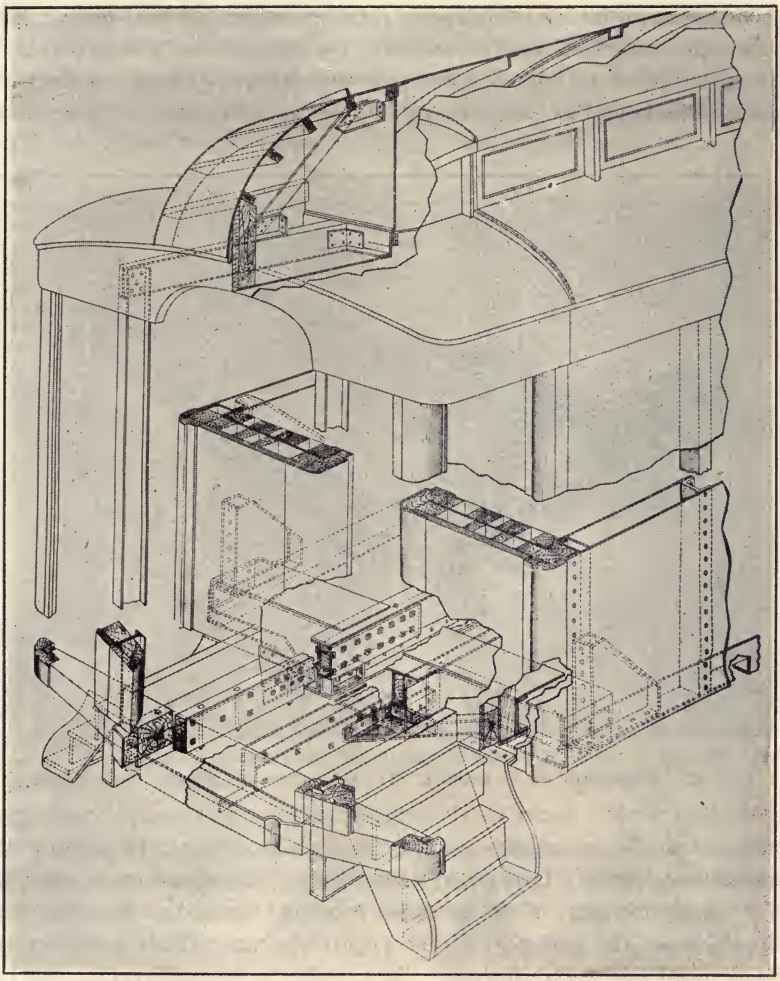

FIG. 14 COLLAPSIBLE VESTIBULE MADE OF A SERIES OF WOODEN POSTS TO SECURE ADVANTAGE OF ELASTIC AND CUSHIONING PROPERTIES OF WOOD

much greater than a car of the same size of wooden construction, and that the wooden car possesses in itself a natural elasticity to absorb

buffing shocks such as are produced by collision that the steel car does not furnish. Hence, in the development of the steel car, with the enormous increase in weight of trains and the high speed at which they run, there has been a growing tendency to increase the strength of the structure with the view of making it as nearly indestructible as possible in order to compensate for the absence of elasticity. It is also apparent that, notwithstanding the strength of the structure, if it encountered an opposing force of sufficient magnitude, it might be annihilated, and so this strengthening process, and the increasing

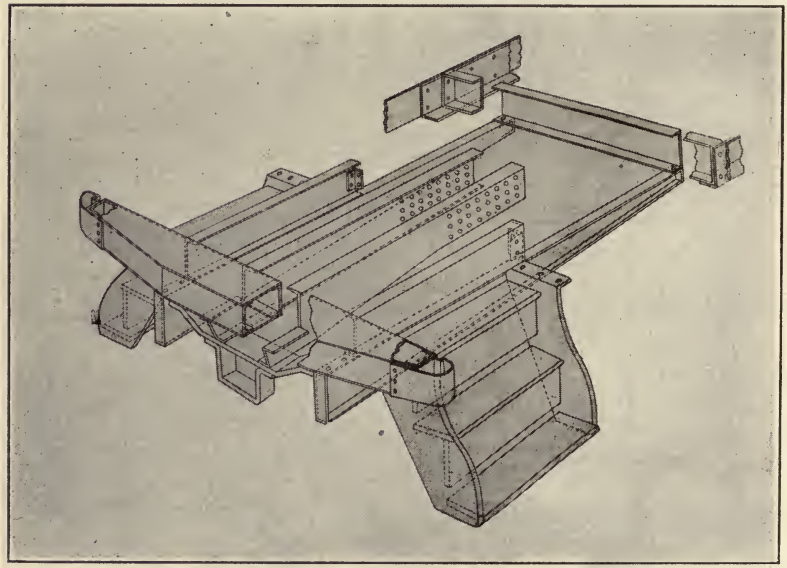

FIG. 15 SKELETON OF PLATFORM MEMBERS FOR ALL-STEEL CONSTRUCTION

weight and speed might go on indefinitely without furnishing the result sought for. It is equally true that if the structure is designed for such strength as to be indestructible, when the two opposing forces meet, the movable objects within the cars, which is the human load, must suffer the damage. To avoid this possibility the idea has been evolved to construct that portion of the end of the car between the end of the main body and the vestibule face plates, these members being all such parts as are embraced in the platform, vestibule and hood covering the vestibule, so that it will collapse under a less shock than would be required to crush in the end of the car body itself.

19 This idea is based on the theory that in a train in which there are say ten vestibuled cars, there is the space between the main bodies of each two coupled cars occupied by the platforms and vestibules of approximately 8 ft., or in a ten-car train a space of approximately 80 ft., of shock absorbing space, which, if properly utilized in the instant of collision, would remove to a large degree the shock and resultant damage to the car body itself and likewise lessen the possibility of damage to the persons of the passengers. From this idea has developed what is termed a collapsible vestibule. It is generaly conceded that if two vestibuled cars coupled together could maintain their

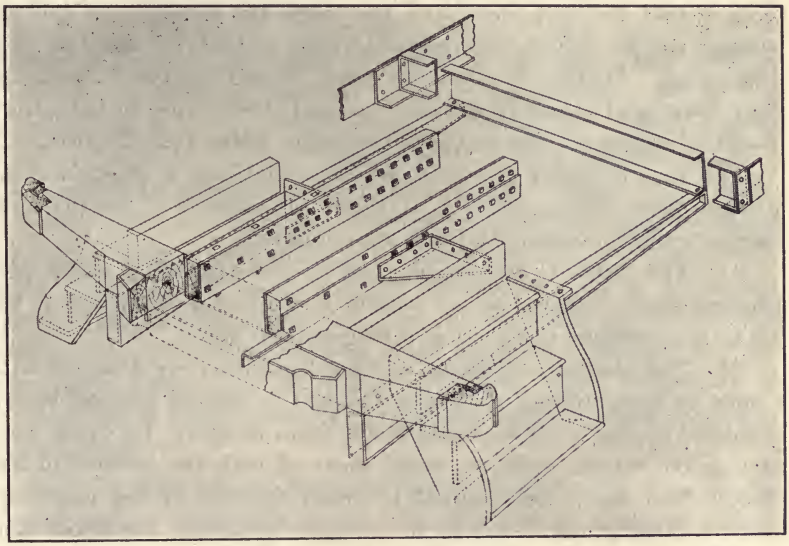

FIG. 16 SKELETON OF PLATFORM MEMBERS, STEEL AND WOOD CONSTRUCTION

respective horizontal planes at the instant of shock due to collision, there could be no telescoping and that telescoping is due to one car assuming, at the instant of collision, a higher or lower horizontal plane than its adjoining neighbor, causing one to ride the other with the resultant telescoping effects.

20 It is generally conceded, that in cases of two cars tending to telescope, the point of maximum shock is never over 20 in. above the floor line. In the Government postal car specifications, this point has been definitely fixed at 18 in. above the floor line, and with this in view the end posts are reinforced for a distance of about 4 ft. above the floor line by steel angles riveted to the Z-bar end posts.

21 A general idea of this collapsible vestibule is afforded by Figs. 13 and 14. Fig. 13 shows the construction entirely of steel, while Fig. 14 shows a series of wooden posts and platform and vestibule members in addition to the steel members to secure the recognized advantage of the elastic and cushioning properties of the wood.

22 In this construction the longitudinal sills and floor members are designed to stop at the end sill of the car body proper, the end of which is sheathed with a heavy steel plate extending in one piece vertically from the roof downward to the bottom of the end sill. If the shock of collision is not entirely absorbed by the vestibule members before the end of the car body proper can be crushed, this plate will tend to pull the roof downward and cause the direction of the oncoming car to deflect obliquely upwards instead of the two cars telescoping. Further to offset the effect, should the two cars change their horizontal planes in collision, pressed steel shapes in the nature of anti-climbers are placed below the buffer beam and platform.

23 Fig. 15 shows the skeleton of the platform members for the all-steel construction, and Fig. 16 shows the skeleton of the platform members where wooden features are employed.

24 The vestibule diaphragm posts are constructed of heavy steel I-beams rigidly secured at the bottom to the buffer beam and at the top to the vestibule end plate and longitudinal braces.

25 The platform, vestibule and hood members are designed with a view to withstanding all shocks incident to regular service, but in abnormal shocks, such as would result from collision, the rivets connecting the various members would shear off with the exertion of less energy than would be required to crush the end of the car body, thereby causing the vestibule to collapse, absorbing the shock and furnishing a cushion between the two car bodies proper. It is assumed that in case of a collision these would be the only parts seriously damaged, and the car could be repaired and replaced in service with a minimum of expense and delay.

26 The entire collapsible vestibule, comprising the platform, vestibule and hood, is constructed as a unit, detachable and separate from the car body proper and can be applied after the car is built or in the alteration of cars already built and is equally applicable to cars of either steel or wood construction.

27 The object of the collapsible vestibule is, first, to protect the lives of the passengers and secondly to protect the body proper of the car from serious damage.

No. 1388 *n*

DISCUSSION ON STEEL PASSENGER CAR DESIGN

GEORGE GIBBS. On this occasion when the subject of steel passenger car design is under discussion, it may be of interest to make a brief reference to the early history of this important innovation in railway operation which had its origin in connection with the provision of car equipment for the first rapid transit subway in the City of New York. The writer was at that time consulting engineer of the subway construction company in charge of car design and construction. It was obvious that the exacting requirements of the contemplated service, which involved tunnel operation at a high schedule speed with closely spaced trains crowded with passengers, must be conducted with all possible precaution against accident and, further, in a way such as to minimize the fatal consequences of any accident which might occur in spite of such precautions. The two consequences most to be feared from an accident are the breaking up of the cars in the train and the setting of a fire in the wreckage; on an open railway line the consequences of these are serious enough, but in a subway or tunnel they are potentially much worse, because of the confined space which prevents the prompt escape of passengers from the wreckage.

Cars for such service, therefore, should be protected in an unusual degree against the possibility of telescoping in an accident, and the electric apparatus should be installed in such a way as absolutely to prevent the setting of fires. Incombustible metal cars were naturally suggested as the solution of the problem, but in the latter part of 1901, when the question of car design was taken up for the subway, it seemed impracticable to consider the adoption of an all-steel car for the large amount of equipment required, because of the fact that no practical steel passenger cars had ever been constructed and it was evident that to develop a serviceable type a number of very serious mechanical problems had to be attacked by thorough study and experimentation. Not least among the problems was that of keeping the weight of a metal car within reasonable bounds, without sacrificing its strength and serviceability, light weight being an essential requirement in rapid transit operation.

As the best practical solution of the car question, it was, therefore, decided to provide wooden cars initially, but to make them of an advanced type with metal underframes, protected floors and copper-sheathed sides, and to mount the electric apparatus in incombustible envelopes. These cars were rightly considered at the time a great advance upon previous practice in safeguarding against accidental fires. The first lot of 500 of these protected wooden type subway cars was ordered in December 1902.

While it was necessary to insure the operation of the subway at the date set for its opening by providing the initial car equipment, the writer believed the steel car feasible, and in this view he was encouraged by George Westinghouse, under whose stimulating advice he was led to persevere in efforts to develop a metal car at the earliest possible date. A. J. Cassatt, president of the Pennsylvania Railroad, was also impressed with the necessity for non-combustible cars in tunnel service, as the great project of the Pennsylvania road in building a tunnel entrance into New York City was then in progress. He accordingly offered to the subway company the facilities of the Altoona shops to build, in the quickest possible time, a sample steel car, the design of which the writer had completed in October 1902. August Belmont, president of the Rapid Transit Subway Construction Company, concurred in this arrangement and early in 1903 the Altoona shops started upon the construction of this car, which was completed in December of that year. Realizing the many difficulties which would be encountered in getting material promptly at that time, commercial shapes were quite generally used in the design, and the car as built was found, therefore, to be quite heavy and not altogether sightly in appearance.

The company still needed 300 cars to complete the early requirements of the subway operation and it became a question of immediate necessity to determine whether these cars should be of wood or be of the all-steel construction. The writer was able, from experience with the sample car, to develop a new design and at a meeting of the executive committee of the subway construction company early in 1904, he definitely recommended the letting of contract for 200 of the new design of steel cars. On the strong endorsement of E. P. Bryan, general manager of the road, Mr. Belmont decided to venture upon this important innovation in railroad operation. The contract for the 200 steel cars was accordingly let in March 1904, and followed in October of the same year by 100 more. Both these contracts were taken by the American Car & Foundry Company, which had the

courage of its convictions in assuming the heavy responsibility of turning out these large orders at specified time and at a price which was not out of line with that of the previous wooden cars. A number of these cars were received in time for the opening of the subway, October 27, 1904, and are running today.

During the same year the writer, who also had charge of the electrification of the Long Island Railroad, placed an order for 122 steel electric motor cars of practically the same design as the subway cars; this service started June 28, 1905. The Long Island was the first steam railroad in the country to adopt steel cars for its passenger service.

The New York Central a year later placed an order for 125 steel cars and inaugurated their electric service from the Grand Central Station in January 1907.

The Pennsylvania Railroad, as a result of the progressive action of Mr. Cassatt, endorsed by Samuel Rea, then vice-president of the company, adopted steel passenger cars for all trains coming into the new terminal, a decision which has since had a far-reaching effect upon the standards of all railways of the country. The question of the best design for through passenger train cars was taken up exhaustively and systematically by this company and was made the subject of a report by a special committee of its operating officials in May 1909. Today this company has in service 2139 steel passenger cars, excluding a large number of sleeping and parlor cars of the Pullman Company, and builds no other type.

William F. Kiesel, Jr. The method of suspension described by Mr. Summers may be very good on short cars, but with long cars, especially passenger cars, it does not seem sufficiently flexible in the trucks to avoid unbalancing and putting the cars out of shape. The bodies are long and the cars have some flexibility, but in some cases the tracks are such that it is necessary to have excessive provisions for flexibility aside from that in the truck.

John A. Pilcher. Referring to Par. 2 of Mr. Summers' paper, the question of the amount of wind that has to be taken up between the two trucks on the car seems to be exaggerated; in approaching a curve the rise in elevation of the outer rail is about 1 in. in 50 ft. On the ordinary modern passenger car truck centers are about 50 ft. apart so that the total amount of wind is about 1 in. measured at the rails. Considering the car weighs 130,000 lb., with trucks approximately

20,000 lb. each, and the car body about 90,000 lb., in order to take care of the wind in the track, the springs on the diagonals of the car would have to compress ½ in. more than the springs on the opposite diagonal assuming the springs as over the rails. On this same car this would mean that two diagonals would have 20,150 lb., and the opposite two 24,850 lb., or a difference of 4700 lb. This difference in deflection is taken from actual springs.

To analyze this in connection with the swinging hangers, assume these hangers to be 11 in. long, and to be located at an angle with a vertical of 28 deg. 8 min., which is about that shown in the cut, and also assume that they are located approximately over the track (the movement would have to be decreased or increased in proportion to their distance from the rail inside or outside) with a load of 22,500 lb. for each group of hangers.

In order to take care of the same amount of wind in the track as considered in connection with the springs, that is, ½ in. difference in elevation on the opposite sides of the track, the angle would be decreased to 25 deg. 35 min. on one side, and increased to approximately 30 deg. 45 min. on the opposite side in order to bring about stable equilibrium. The vertical loads would amount to 24,935 lb. on one side, and 20,065 lb. on the opposite side, or a difference of 4870 lb., just a little more than when the springs were used. In calculating the deflection of the springs only that of the bolster springs was taken into consideration; the equalizer springs would also have to take an additional load, and would help to reduce the difference of loads necessary to bring about the proper amount of deflection.

Looking at a car from the rear approaching a curve, when the front truck enters the curve the centrifugal force at that point would tend to throw the car body, relative to the truck, actually in the opposite direction from what it should move in order to equalize the stresses. This would put additional torque in the body of the car, which would not be present in the case when springs only take care of this movement. The torque would be rather reduced at the time of entering the curve when the springs only are used.

When both trucks are on the curve all of the wind is out of the car; the centrifugal force in that case throws the car body toward the outside, and would tend to augment the lift in the track on the outside, which is hardly desirable.

Angular hangers, while they may not have been intended for the purpose described, have been in use for a number of years. It is very questionable whether they are of any advantage.

SYMPOSIUM ON FIRE PROTECTION

No. 1393 *a*

DEBARMENT OF CITY CONFLAGRATIONS

By Albert Blauvelt, Chicago, Ill.

Member of the Society

Opinions take opposite sides as to whether the central districts of our leading cities are today subject to conflagration.

2 Passing over interested and private opinions, the 1911 report of the Joint Committee of the Senate and Assembly of the State of New York, on insurance and fire waste; the Illinois Fire Insurance Commission Report of 1911; the Ohio Fire Marshal Report for 1907; the United States Department of the Interior, Bulletin 418, of 1910; the Wisconsin Senate and Assembly Committee of 1913 and all other public reports the writer can find agree that each of our cities is today subject to conflagration. Our cities appear to spend in public and private ways enough money to make a conflagration impossible for the central districts, but they do not achieve this result.

3 The horizontal hot blast of a heavy fire driven by wind ranges from a few hundred to more than a thousand feet, whereas the most powerful hose streams are not effective at over one hundred to two hundred feet.

4 Such fires develop in the heart of our cities in the face of more firemen and apparatus than can be quickly used, as at Boston in 1872 and 1889; Montreal in 1901; Paterson in 1902; Baltimore and Toronto in 1904; and such fires also develop in city outskirts or cheap quarters and by force of wind sweep into the city proper as at Portland, Me., in 1866; Chicago in 1871; St. John, N. B., in 1877; Ottawa-Hull in 1900; Jacksonville, Florida, in 1901; at Chelsea, Mass., and other cities.

5 Inasmuch as any engineering plan to debar conflagration, if intended to cover an entire city, inclusive of cheap districts, would be prohibitive in cost; any such plan must be limited to the central or high value district and designed to prevent free spread of fire

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within same, and also debar any deep inroad of conflagration from without such district.

6 Every conflagration must necessarily begin in one of two ways, or a combination of both, Chicago in 1871 and Baltimore in 1904 being examples of the two types.

7 The Chicago fire started outside of the congested district, developed into hot-blast form, and swept through and beyond the congested district, and burned out for lack of fuel.

8 On the other hand, the Baltimore fire began in the heart of the city and ramified more swiftly than the firemen could operate; then took the hot blast form and burned out for lack of fuel.

9 Such a hot blast has never been stopped by firemen while the wind held, but has, however, been checked and deflected upward by barriers consisting of two or more fire walls or their equivalent, with a free air space between, as per the known instances of fires out of control, which have been stopped by a mere alley if fully shuttered on each side.

10 It has also been possible to absorb the hot-blast attack of such fires by a very deep and fixed mass of spray in the form of sprinklered buildings. The Boston fire of 1893 was largely absorbed by an exceptionally good water supply in such form, and somewhat similar experiences have been had at Toronto in 1904, and other cases.

11 These successful experiences in checking hot-blast fires by deflecting the flame or by absorbing it in a mass of spray have been but little appreciated and instead of acting upon the lessons which they teach, our cities today have a collection of safeguards, part of which lend themselves to the debarment of conflagrations and part of which do not.

SAFEGUARDS WHICH SINGLY CANNOT DEBAR CONFLAGRATIONS

12 The recognized and partly recognized safeguards against fire, no one of which alone can debar conflagrations, are twelve in number, viz., fire prevention; the fire limits; the water supply; the fire department; the high-pressure fire system; the uniform hose thread; the water curtain; the so-called fireproof building; the horizontally divided building; the protected window; the sprinklered building; and the piped building.

13 Dynamite, private hose, steam jets, carbonic gas systems, and fire walls separate from buildings are not listed because they are not recognized by fire chiefs for valid reasons.

14 *Fire Prevention.* The preponderance of disasters from trivial, unknown or unguessable causes appears to forbid hope of elimination of conflagrations through fire prevention. A half century of experience also shows that the skill and effort directed to prevent city fires from becoming disasters has been successful within 0.00003 of the total fires. This 0.00003 is what has hurt, and appears to be the only considerable task of correction remaining for the engineer.

15 *The Fire Limits.* This is an expression indicating a central territory, within which frame construction or shingle roofs are prohibited and better construction enforced, especially for large area buildings. The elimination of frame buildings and shingles is an essential part of any plan to debar conflagration.

16 *The Water Supply.* The water supply is indispensable and also adequate in most of our cities. Were this not so, Baltimore and other conflagrations would have gone to the extent of San Francisco, whose water mains were wrecked by earthquake.

17 *The Fire Department.* While no fire department has ever been able to put water on the front or rear of any hot-blast type of conflagration, nevertheless, the fire departments at time of conflagration have been of gigantic value in keeping the fire from spreading across the wind, in extinguishing brands thrown far ahead, etc. The fire department, therefore, is indispensable and this paper argues to increase its opportunity, but not its cost and size.

18 *The High-Pressure System.* Those cities which are partly equipped with costly high-pressure systems enjoy an advertisement which does not appear to be shared nor courted by the group of valley cities with reservoirs on high bluffs or hills. The latter afford a high-pressure service for the whole, not parts of such cities. Powerful high-pressure hydrant systems have also long existed in a goodly number of cities in the form of special inland hydrant lines operated by fire boats.

19 High-pressure develops long and large hose streams, but there is nothing about it to enable firemen to use such hose streams either in advance, or in the rear of, a hot-blast conflagration, nor is a high-pressure system especially flexible to check the ramification of fire as at Baltimore in 1904.

20 *The Uniform Hose Thread.* Since the fire department is indispensable, ability to double up departments is obviously wise and Mr. Griswold's long labors and the report of the sub-committee on fire protection on this subject are exactly in point.

21 *The Water Curtain.* It is fair to say that a water curtain

has successfully held off some heavy fires, but never a conflagration. The fatal weakness of the water curtain is that it is blown away or scattered by the brisk breeze which necessarily accompanies a hot-blast conflagration. The depth of the spray is also far too shallow.

22 The water curtain, as shown by fire record results, is a valuable safeguard for moderate exposure fires, or even for fairly severe exposure when favored by quiet air, but fails in severe tests.

23 *The Fireproof Building.* The typical, so-called "fireproof" building, having merely incombustible floors, roof and walls, cannot debar a conflagration (because of its unprotected windows and large volume of contents) any more than can an ordinary brick building with a good roof.

24 A hot-blast conflagration moves laterally, and a "fireproof" building in its path, as evidenced at Baltimore, is merely a crate which holds up the fuel contents in position for free burning and augments the general hot blast.

25 It is contents, not buildings, which make the bulk of property loss; contents, not buildings, which are hazardous; and contents cause buildings to burn in most cases.

26 Repeated experience shows that no building can withstand the heat due to burning any large quantity of contents or even very moderate contents if in a large rotunda, half floor office or like large area. The writer submits to all who are authorities on heat the feasibility of constructing a high structure of large retorts, each capable of restraining 10,000 to 50,000 lb. of ignited fuel, and also filling with customary taste and beauty the needs of utility and health for habitation.

27 The fireproof building has unequalled habitability and utility and also lends itself admirably to conversion into a piped building, and even more excellently lends itself to conversion into a building having protected windows.

28 *The Horizontally Divided Building* is designed to hold a fire from rising through the floors. Repeated experience in tall buildings has proven that excessively large and dangerous fires develop quickly if the floors are not fire tight.

29 Such horizontal division is obviously of no value against a hot-blast fire moving laterally from without, and it also does not prevent an internal fire from jumping from floor to floor on the outside of the building.

30 Horizontal fire-tight construction is very rarely found when

fire-tested, its worst drawback being that in the opinion of the public it injures the habitability and utility of the building, hence all sorts of concessions are made.

31 *The Protected Window.* Protected windows can debar city conflagrations, but do not because there are too few of them, except in a few special and minor localities.

32 *The Piped Building.* The building piped with fusible outlets, whether with automatic water supply or with water promptly applied, can but does not debar city conflagrations for the same reason given respecting the protected window.

TRUE HOT-BLAST CONFLAGRATIONS CAN BE DEFLECTED OR ABSORBED

33 Hot-blast conflagrations have been successfully controlled both by deflection and absorption.

34 Taking up the deflection idea first, all will doubtless agree that if the four walls of every city building were solid brick with no doors or windows, a spread of fire would be impossible, even with no fire department. From this it follows that if all doors and windows were protected by wire glass, shutters or fire curtains, the walls of the buildings would have the same tendency. Any experienced fire chief will testify to the enormous fire-stopping effect of an alley shuttered on both sides, the heat and hot blast being deflected upward.

35 Experience shows, however, that when a hot blast reaches a building prepared to act as a deflector by suitable protection for its doors and windows, the first result is partial failure. This is owing to the fact that the heat will radiate through the wired glass, or leak through the shutters, and ignite the contents.

36 Nevertheless, there will be a retardant effect and it is obvious that if other buildings located to the right or left of the center of the hot blast have window openings, similarly stopped, they must suffer less and the elsewise lateral ramification of the fire decrease.

37 The hot blast is thus largely deflected upward, partly checked and less able to cross the next street or alley, assuming protected windows throughout.

38 Just how many deflector walls and air spaces could be jumped or burned through by a conflagration of given severity is a matter of judgment based on observation, precisely as the extinguishing power of a hose stream is a matter of judgment from experience, not reducible to exact figures.

39 The writer submits that if all the alley windows were pro-

tected and also all the street windows on the second floors and above in the solid three and four-story parts of a town, a conflagration from without could not then bore a hole or a bay into such a district deeper than through four deflector walls and across three air spaces, which would mean two blocks and three streets, of which one might be an alley.

40 Not that the fire would be put out, nor that tongues and fire brands would not have to be taken care of; but that the hot blast would be deflected upward so the firemen could take a front stand and the general advance and ramification of the fire subside to a state of normal fire department control.

DEBARMENT OF CONFLAGRATIONS BY ABSORPTION

41 There remains but one other known means to regain control of a conflagration, that of absorbing the hot blast by means of the piped building. Experience has demonstrated that a hot blast can be absorbed by a spray if the spray be very deep and fairly housed from the wind as is true of the cage of spray represented by a sprinkler installation in full action in a building whose windows have burned out.

42 The most notable demonstration of this was the Brown-Durell sprinklered building at Boston in 1893. Inasmuch as this building became a single large cage of spray which absorbed the main body of a down-town fire that was wholly beyond control, it is certain that a row of such cages of spray, if placed two or more deep, would always accomplish the same thing, and do so without the aid of protected windows.

43 The writer submits that if a city throughout all of its three and four-story and higher parts were composed exclusively of suitably piped buildings, and special water supply provided, a conflagration from a district without could not burn across a street, through a block deep of spray, and across the next street.

44 The fire would not be put out and fire brands would have to be taken care of; but there would be no ramification of fire in the sprinklered territory and there would be a full restoration of normal fire department control.

VALUES, COSTS, GAINS

45 The Boston big fires proved out burnable property values at a rate of over $500,000,000 per square mile, and it is well known that

today there are several city centers which have grown to a far higher rate of concentration of value.

46 It seems fair to assume $250,000,000 per square mile as an average burnable value over the central districts of our twenty leading cities.

47 For such a square mile, standard automatic sprinklers (including masonry) would cost about four per cent of the burnable values, or $10,000,000 with fixed charges of about 16 per cent per year.

48 Empty sprinklers, or, protected windows, would each cost

TABLE 1 FIXED CHARGES, FIRE COST SAVING AND NET GAIN ON A BASIS OF $250,000,000 PER SQUARE MILE

On an Annual Basis	For Standard Automatic Sprinklers with Double Water Supply	For Empty Sprinklers and Piping to be Supplied by the Fire Department	For Shutters, Fire Curtains, or Wired Glass Applied as per Preceding Test
Investment per sq. mi.	$10,000,000	$5,000,000	$5,000,000
Fixed charges as given above	1,600,000	450,000	450,000
Saved by eliminating risk of conflagration at 33 ct. per $100	825,000	825,000	825,000
Saved by eliminating common exposure fires at 7 ct. per $100	175,000	175,000	175,000
Saved by reducing fire cost within buildings in which fire originated	2,000,000	1,125,000	nominal
Difference between charges and savings	1,400,000	1,675,000	550,000
Per cent earned by savings	14	33.5	11

about half as much, or $5,000,000 per square mile for either, and each incur fixed charges (about the same as the buildings), or about 9 per cent.

49 The savings per $100 of burnable values per year, would be as given in Table 1.

50 In surveying any actual square mile it would develop that but one of the three, viz., protected windows, automatic pipes, or empty sprinkler pipes, would best suit any one building, and this would be likely to result in a detail plan calling for gross investment of about three per cent of the burnable values at a net gain of about 18 per cent.

51 But figures cannot include the grief, loss of work and trade following every large conflagration.

52 Our fire limits, fire departments and waterworks are today well developed, and protected windows or piped buildings throughout the costlier districts are all that is needed, to debar conflagrations.

RECAPITULATION

53 To recapitulate the advantages and disadvantages of the protected window and the two types of piped buildings:

54 The protected window delays the entry of severe fires and also prevents general ramification of fire through innumerable window openings.

55 Not that the protected window does this perfectly, because shutters may be out of order or not within reach to close if open, and because wired glass transmits heat by radiation very rapidly. Nevertheless, as aided by existing air spaces, alleys and streets, the protected window is a proven success.

56 The protected window is beginning to be required in building codes; it also is tangible to the public eye, something that can be seen as representing a fire stop or check; in wired glass form it has some working advantages, at least for skylights, and finds favor with architects on the better class of buildings; in the form of shutters, the fire-stop effect is better than for wired glass, but this is largely offset by the fact that shutters do not get the care which comes to a window which is in more or less constant use.

57 The advantage of the piped building with automatic double-source water supply, the well known sprinklered building, is first of all the protection to life. Apparently this specific form of fire protection is the only one which to any dependable degree conserves life. An experience with say 10,000 buildings over a period of about 15 years gives rise to the statement that no life has ever been lost in a building so equipped, either by fire or smoke, and to the best of the writer's knowledge this is literally true.

58 The operation of an automatic sprinkler system develops a powerful drenching spray not only on the fire but around it, and compels escaping smoke to pass through a dense spray which takes up the acrid quality and heavier carbon contents of the smoke, and thus has much to do with the protection to life.

59 While mathematical safety against loss of life is probably impossible, it is within the truth to say that where people are in masses, or are asleep, safety cannot exist if the fire hazards are not under the automatic sprinkler.

60 A second advantage of the automatic sprinkler system, and the one most in point under the title of this paper, is that it has been found in practice that, given brick buildings, well secured pipes, and reasonable water supply, a fire even when of conflagration magnitude cannot burn completely through such spray further than the depth of one, or say two, buildings.

61 A third advantage is that the fires are put out so quickly and with such economy of water by reason of its accurate application, with so little smoke and so great a reduction of the harmful quality of the smoke that the aggregate fire, water, and smoke damage to goods is far less than for any other form of protection.

62 Referring to the plain piped building, or building equipped with fusible sprinklers on empty piping with exterior hose coupling for fire department use and relying solely on the fire department for water supply, the main advantage is that the first cost and low fixed charges make it applicable to the medium value buildings.

63 Another advantage is that of safety to life, compared with that of buildings not piped at all, because in practice the empty sprinklers can be operated nearly as quickly, and necessarily to the same effect as automatic water-supplied sprinklers.

64 Still another advantage is that the technique and upkeep essential to efficiency are far less than with the full standard automatic sprinkler equipment.

65 The main disadvantage of the protected window is that it protects only as between neighboring buildings and this saving averages too small an amount to cover its fixed charges, on a basis of every-day fires.

66 A disadvantage of the standard automatic sprinkler system is that it is a special engineering product, technical to a high degree, yet depending on this quality for its efficiency, an efficiency imparted by a few skilled contractors and experts. The system therefore is open to criticism by all who rail at any control of skill or service.

67 Another objection to this form of piped building is that its water supplies are often direct from city mains through the influence of large property owners, thereby saving them the expense of private water supply. In Manhattan Island and Chicago the city water is of too low a pressure to be so used, but in other cities there are too few sprinklered buildings to check a conflagration and just about enough of them to jeopardize complete crippling of waterworks and fire department at such a time by reason of these buildings being

wrecked and bleeding the general water supply throughout the breaking of large pipes.

68 Hence a wise requirement for a piped district, would be to provide a special border pipe line into which water would be pumped or admitted under control.

69 Still another disadvantage of standard automatic sprinkler equipment is in first cost and in fixed charges. The investment and fixed charges do not have any fairly constant relation to the value of building plus contents, and at city labor costs are usually excessive, except for large and costly buildings.

70 A disadvantage of the empty pipe sprinkler system is that this mode of protection has as yet but few applications; no extended study has been given the art of cheap extinguishment of fire in medium value property. Another disadvantage is that fire department practice is at variance; some chiefs favor and ask for such equipment, and others evade or object.

71 It does not seem to be generally realized that a building in a central district does not burn badly before the department arrives. Were this not so, modern fire departments would not, as the records show year after year, hold the fires within moderate loss, except 0.003 to 0.005 of the total. The fire department does arrive while the fires are yet incipient, though perhaps inaccessible. It seems to be accepted as a matter of course that a costly proportion of buildings shall burn and soak, subsequent to the arrival of the department, for the sole reason that the department cannot quickly put ample water where, and only where, it is needed. Yet to do the latter is all that the standard automatic sprinkler equipment does or professes to do, and water can be supplied to empty pipes nearly as quickly by firemen as by a private tank.

72 Even in a case of purposely delayed alarm and sprinklers shut off (incendiary), the writer has seen work done in this manner by only one steamer with wonderful success, extinguishing a four-story fire which otherwise would have required many hose streams, and this after there was no time to set up ladders and place hose.

73 The fire cost of empty sprinkler equipment would admitttedly be greater than for automatic sprinkler equipment because while there would be no failures through pipes frozen or valves shut off, the fire department would not put water on the fires at quite as early a stage of incipiency.

74 A willing fire department, however, would put water on the

fire through such pipes while a fire was yet incipient, because our fire department records show that the department arrives and the vast majority of city fires are put out while incipient.

75 To pay for the greater fire and water damage in practice with empty pipes as compared with automatic sprinkler equipment, the figures given in Table 1 allow for the typical square mile, $825,000 per year, to say nothing of $1,160,000 reduction of annual fixed charges.

76 However, the standard automatic sprinkler system has been fully demonstrated for over 20 years, yet it is but just coming into its own, and the empty sprinkler system must in turn wait for recognition and extend in application by degrees.

77 The practical difficulty, therefore, of debarring disastrous conflagrations in our cities seems not in lack of means, nor in lack of knowledge based on experience, so much as in the lack of agreement on a plan, and the difficulty of apportionment of expense.

No. 1393 *b*

BALTIMORE HIGH-PRESSURE FIRE SERVICE

By James B. Scott, Baltimore, Md.

Member of the Society

The Conflagration of 1904. The conflagration of 1904 was due to the simultaneous occurrence of an incipient warehouse fire gaining headway unobserved on a Sunday morning; a high wind; inferior building construction and inadequate fire-fighting equipment.

2 During the first eight hours of the fire the wind was blowing from the southwest, after which it shifted 90 deg. to the northwest, causing the fire to advance with its broadside of 1500 ft. for a front. Although supplemented by engines from other cities, after the fire had got beyond control the operations of the department might be described as a skilful retreat, an engine and a truck being lost under falling walls because the retreat had not been sufficiently rapid. Dynamite was freely used by skilful operators, but was practically ineffective. The fire raged for 30 hours, covering 150 acres, causing a loss of $100,000,000 and finally burned itself out when the wind changed again to the north and drove the flames toward the open harbor.

3 There was no scarcity of the water supply. The topography of Baltimore shows elevations ranging from 6 to 460 ft. above mean low tide, and to prevent excessive pressures in the low lying sections or a deficiency in those higher up, the supply is divided into five separate services. The "low" and "middle" services are fed entirely by gravity, the three higher services being supplied by pumps and high storage reservoirs. Suitable by-passes are provided so that in an emergency any service can be supplied from the next higher. At the time of the conflagration there was available a total reservoir capacity of over 1,750,000,000 gal., in addition to pumps of 63,000,000 gal. capacity. At that time the consumption for domestic and industrial purposes was about 60,000,000 gal. daily, and the draft from the reservoirs for fighting the conflagration was approximately the same amount, or a trifle over 3 per cent of the available reservoir

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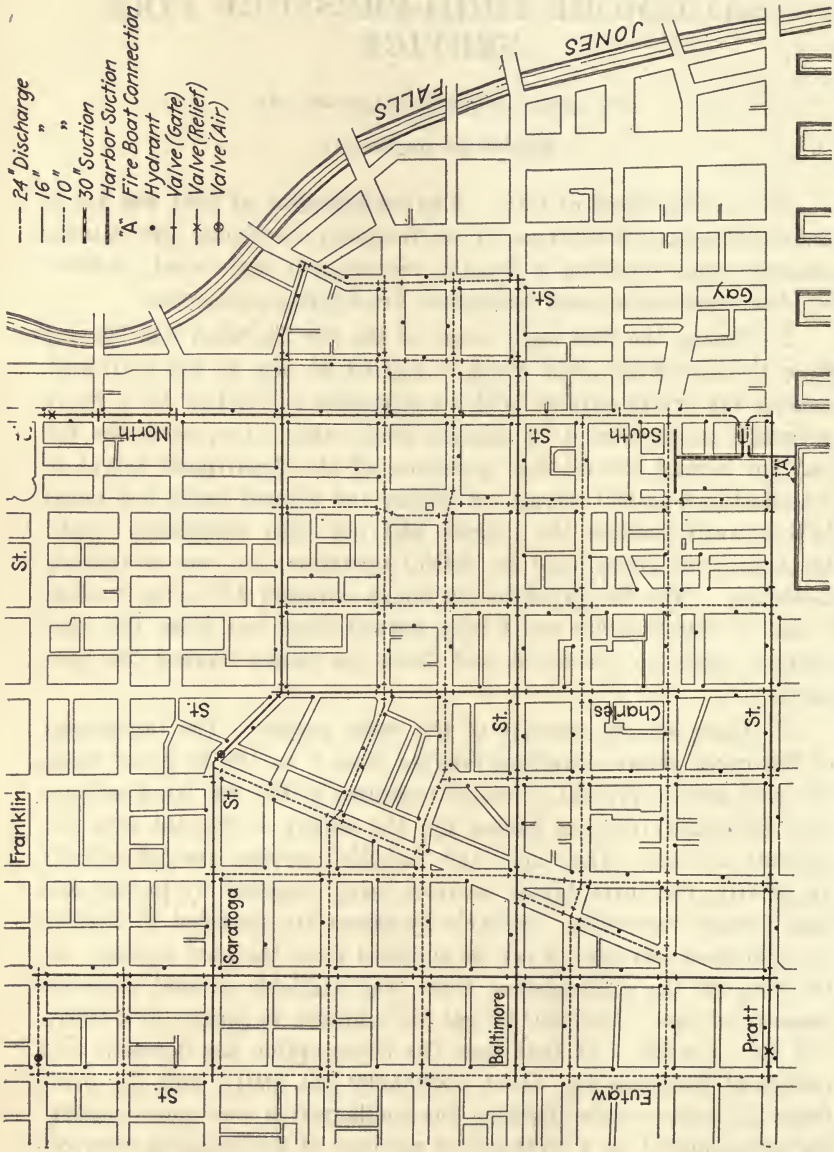


FIG. 1 MAP OF THE HIGH-PRESSURE DISTRICT

capacity. In addition to the reservoir draft, about two hours after the fire started the two 17,500,000-gal. pumps in one of the high-service pumping stations were by-passed into the "middle" or upper gravity service. The fire originated in the middle service district and shortly afterward extended into the low service district. As soon as this occurred the by-pass between the two was opened, and the pressure was maintained at 80 lb. during the remainder of the fire.

4 The operations against the conflagration demonstrated that, for effective fighting of a dangerous fire, large volumes of water must be delivered on the fire with the least possible delay, and at suitable pressures. To meet these requirements a separate high-pressure fire service was designed and installed, covering the greater portion of the congested value districts of the city.

5 *Characteristics of the Congested Value Districts.* The corporate limits of Baltimore embrace an area of 31½ sq. mi., with a density of population of 17,729 per sq. mi., a density greater than any other of the larger American cities. The congested value districts cover approximately 300 acres, the area at present covered by the high-pressure mains is about 175 acres, or assuming that the territory for 200 ft. outside the mains is protected, the high-pressure district may be considered to be 245 acres.

6 The elevations of the congested value districts vary from 6 ft. to 100 ft. above mean low tide, about one half of the area being below elevation 50. The pump centers at the high-pressure station are at about elevation 12.5.

7 The character of the building construction in the congested value districts varies from first-class modern fireproof structures to converted residences, except in the rebuilt burned district, where a better type of construction exists. In the districts in question all electric wires have been placed underground in conduits owned by the city, with the exception of the street railway trolleys. Before rebuilding the burned district, several narrow and congested streets were widened and plazas established around important public buildings, so that the conflagration hazard in that section has been considerably decreased.

DESIGN OF THE HIGH-PRESSURE SYSTEM

8 *The Time Element.* High-speed automobile hose wagons are provided, housed at convenient locations, in order that the minimum

time possible may be consumed in reaching the scene of a fire after the alarm is received. Each wagon is equipped with a four-cylinder four-cycle motor of sufficient power to develop a speed of 30 mi. an hr. through the streets of the city, with a load of 5000 lb. Each body is 50 in. by 34 in. by 12 ft. inside, and carries 2000 ft. of 3-in. hose. Each body is equipped with one 2000-gal. Morse Invincible monitor nozzle and two 1100-gal. monitors.

9 *The Pressure Element.* The pressure required for effective work varies widely according as conditions demand the flooding out of a basement fire or fighting on the top floor of a modern skyscraper. At times both extremes may be required simultaneously.

10 To meet efficiently these conditions requires the maximum pressure to be available at each hydrant, with means for the separate control of the pressure on each hose line. The specifications called for a combination operating valve and regulator capable of adjustment from shut-off to 50, 75, 100, 125, or 150 lb., or if desired, to the full line pressure of 300 lb. The pressures were to be plainly marked on the valve by notches to be used by the operator as a guide for setting the handle. Regulators were to hold the pressures steadily within 10 lb. of the set amount, whether the play pipe were open or closed. The regulator was also to be provided with a lock whereby the handle could be prevented from passing the 150 lb. notch, but after unlocking could be moved to the full line position. When wide open under the maximum pressure of 300 lb. the valve was not to show a loss of head in excess of 15 lb. The hydrant head was to contain four horizontal outlets and one vertical outlet, each horizontal outlet to be provided with a regulator, and the complete head including four regulators was not to weigh over 110 lb. The head and regulators were to be designed for 300 lb. working pressure and were to be tested with a static pressure of 600 lb.

11 While the above requirements may seem simple enough, only one of the valves submitted by different makers met the requirements in all essential points, a regulator designed especially for the purpose by the Ross Valve Manufacturing Company of Troy, N. Y. (Figs. 2 and 3).

12 The main regulating valve is inserted in an opening just over the hose connection, and is inclined outward at an angle of 20 deg. The opening is closed with a plate carrying a pilot valve and a guide for the main valve. The pilot valve with its diaphragm is covered with a spring chamber, the whole being held in place by cap screws.

The main valve is balanced and is provided with a flat seat and leather face. The upper part of the main valve acts as an operating piston, being provided with a cup leather packing. The pilot valve is balanced against the delivery pressure by the regulator springs, which are made double to secure a wide range of pressure in a short length. The top of the spring chamber is revolved by the operating handle attached to it, and being provided with a coarse square thread screw, less than one revolution is sufficient to give the full range of pressure on the springs from full open to closed. The pilot valve is held positively in the two extreme positions independently of the springs by stops at the top and bottom of the stem. The full hydrant pressure is admitted to the operating chamber of the main valve through a small tube projecting below the seat of the valve. This tube is extended in order to keep the entrance clear of the varying velocity near the valve seat, which would tend to vary the flow of water to the operating chamber. When the pilot valve is open, water wastes from the operating chamber, the pressure is lowered and the main valve is opened by the unbalanced pressure below. When the pilot valve is closed the full hydrant pressure is maintained in the operating chamber, and as the area of the operating piston is somewhat larger than the area of the main valve, the pressure is unbalanced in the opposite direction and the main valve is closed. Intermediate positions of the pilot valve are followed by corresponding movements of the main valve, maintaining the delivery pressure within a few pounds of the amount indicated by the notch at which the operating handle is set. The main valve when fully opened presents an unobstructed waterway. The entire mechanism is simple in design, easy to operate and has proved entirely satisfactory in service.

13 *Quantity of Water Available.* The National Board of Fire Underwriters after a careful study of the situation recommended that a total delivery of 15,000 gal. per min. should be available within any area not exceeding 100,000 sq. ft., at a pressure of not less than 200 lb. at the hydrant. As installed, each hydrant has four 2½-in. horizontal and one vertical outlet for mounting a monitor nozzle. At present the 3-in. hose is connected through reducer couplings, but in the future all hydrants will be equipped with 3-in. outlets and connections to 2½-in. hose will be made through reducers.

14 The quantity of water which can be delivered through a hydrant is a function of the number, length and size of hose lines

attached, and the diameter and type of nozzles. Assuming four lines of 3-in. rubber-lined hose each 100 ft. in length, and 200 lb. at the hydrant, with 1¾-in. smooth nozzles, each hydrant would deliver 3800 gal. per min., or the requirement of the underwriters would be met by four hydrants. At least twice that number of hydrants are available for every unit of area mentioned. If 2-in. nozzles were used, other conditions remaining the same, the discharge would be 4400 gal. per min. at a pressure of 86 lb. at the base of the play pipe, as compared with 108 lb. in the former case.

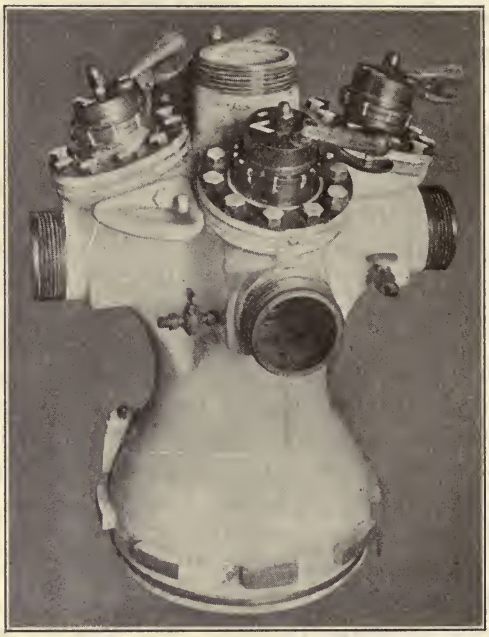


FIG. 2 PORTABLE HEAD

15 At present 226 hydrants have been installed, staggered on opposite sides of the streets, averaging about 170 ft. lineal spacing. In addition to the normal locations at street and alley intersections, the fire chief placed a number of hydrants at special positions, to meet local conditions of exposure, extra hazard or specially congested values.

16 *Design of Hydrant.* The type of hydrant to be used was given careful study. It was considered especially desirable in view of the experience gained in fighting the conflagration, to be able to

place a monitor or special flat nozzle directly on the top of the hydrant, to form a water curtain for the protection of exposed property on the opposite side of the street. Street intersections form a specially desirable location for such a purpose, but usually the corner of the footway is preëmpted for various other structures such as sewer inlets, lamp posts, trolley poles, police and fire-alarm boxes, etc. It was soon evident that if restricted to the ordinary post type of hydrant, it would be impossible to secure suitable locations, and the

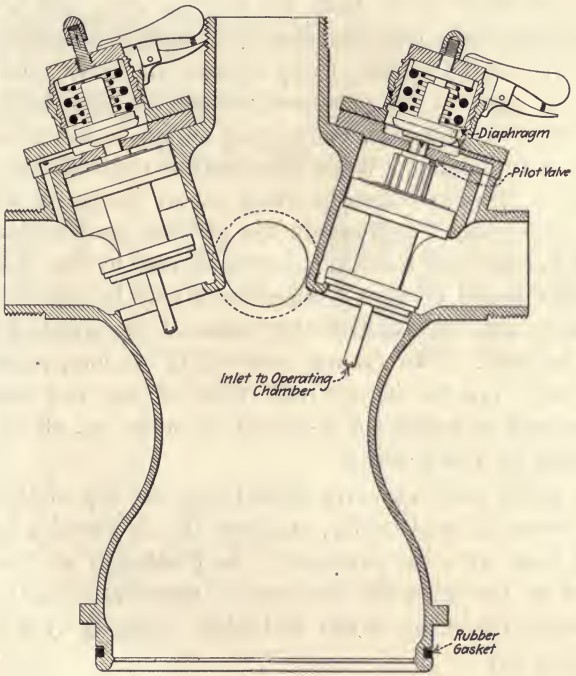


FIG. 3 SECTION OF PORTABLE HEAD

water-curtain feature would have to be abandoned. The flush hydrant with portable head seemed to meet all the conditions admirably. By the use of this type, hydrants could be located almost without restrictions, either in the driveway or any part of the footway. The portable head was also admirably adapted to the use of the regulating valves on the hydrant, as the entire operating mechanism, other than the main valve, could be kept in the firemen's quarters, instead of being exposed to frost or other injuries on the street.

17 The portable head as designed complete with four regulating valves, weighs 110 lb. In order to provide against any delay in attaching the head to the hydrant, a special "bayonet" joint was designed (Fig. 2). The head slips loosely into the barrel of the hydrant, and by a twist of 22½ deg., a series of interlocking lugs on the head and barrel engage each other, and the full section of these bronze lugs is in shear to resist the water pressure. The water joint is made by means of a square soft rubber packing ring placed in a square groove on the outside of the lower portion of the head. The groove is somewhat larger than the packing ring, and at very low pressures water leaks past the ring. At higher pressures, however, the water presses the rubber closely against the barrel, and the joint is absolutely tight at all pressures between 20 and 1000 lb. The action is entirely automatic, there being no screws nor glands of any kind to be manipulated. When the water is shut off, the ring contracts and the head can then be lifted out of the barrel without the slightest resistance. To illustrate the extreme simplicity of the device, and the ease of handling, a recent test by the engineers of the National Board of Fire Underwriters may be cited. The head and operating key were laid in the center of the street, 20 ft. away from the hydrant. Two firemen selected at random, picked up the head and key, ran to the hydrant, removed the two loose covers, placed the head in position and turned the water on, all in the space of 18 seconds by a stop watch.

18 A small cast-iron cover is laid over the top of the barrel to protect it from dirt and injury, and over this is placed a larger cast-iron cover flush with the pavement. As practically all the hydrants are located on the sidewalks the cover is made quite light, and if it should become frozen in, it can be broken instantly by a blow from the operating key.

19 The hydrant proper is designed for a clear waterway of 28 sq. in. through the main valve. The main valve closes with the pressure, and an auxiliary is provided, actuated by the main valve stem. This auxiliary valve opens in advance of and equalizes the pressure before the main valve starts to open. A drip valve is arranged so that as the main valve opens the drip is closed and vice versa, but both valves cannot be open at the same time. Both the main and auxiliary valves have conical leather faces and bronze seats. The barrel of the hydrant is of soft grey iron, and all nuts and fittings, stuffing-boxes, etc., are of bronze. The main valve stem is Tobin bronze, and the

operating spindle is of forged steel. All pressure parts are designed for a working pressure of 300 lb. with a factor of safety of twelve. After erection a field test of 600 lb. was made.

HIGH-PRESSURE PIPE LINES

20 *General Plan.* The general plan of the system is a gridiron of

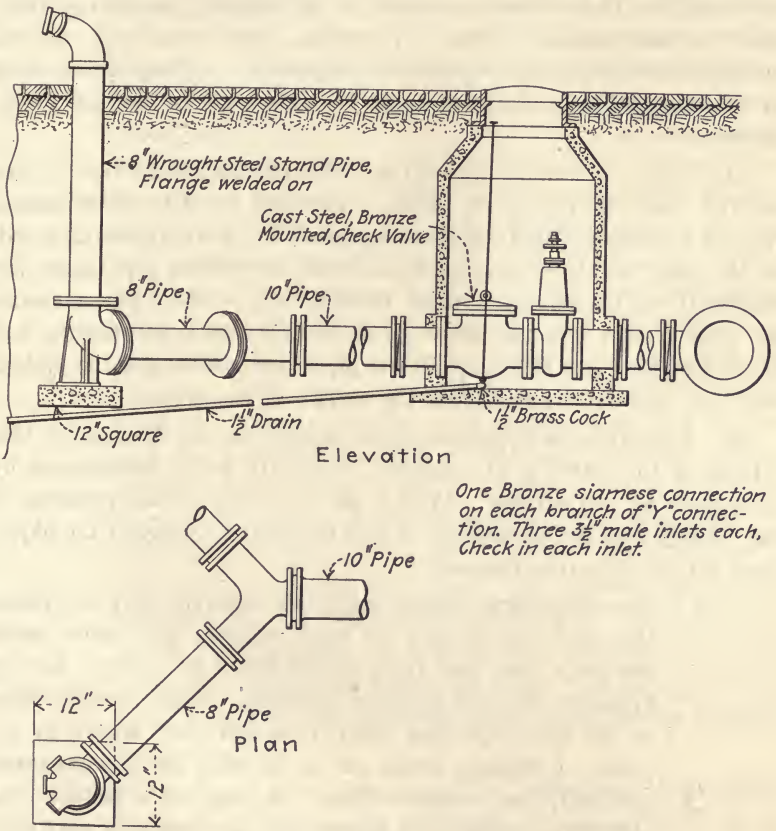


FIG. 4 FIREBOAT CONNECTION

16-in. mains crossing at intervals of approximately 1200 ft. in each direction, with 10-in laterals in the intermediate streets spaced approximately 400 ft. apart. A 10-in. branch is provided at the harbor front for connection with the fireboats (Fig. 4). All hydrant branches are 8 in. There are no dead ends on mains or laterals in the entire system, with the exception of an extension on one street

which will be connected up at the next addition to the system. From the pumping station two 24-in. mains discharge into the intersections of the 16-in. mains. In the pumping station the 24-in. mains are looped around the rear of the pump foundations, thereby avoiding dead ends and by equalizing the stresses render unnecessary any heavy anchorages. Individual 14-in. pump discharges deliver directly into the 24-in. mains through welded necks. Three 24-in. valves and an 18-in. cross connection in the station, serve to sectionalize the large main, so that any portion may be cut out without putting more than one pump out of commission. A large rolled steel air chamber 30 in. in diameter by 20 ft. high is placed on each 24-in. discharge main.

21 *Cast-Iron versus Steel Pipe.* For the service in view it was believed that cast-iron pipe, usually employed for city water-mains, was not a scientific application of material for the stresses involved. On the other hand, commercial rolled steel lap-welded pipe meets the structural conditions much more satisfactorily. With this material the pipe system becomes an engineering structure, all tensile and bending stresses are taken up by the pipes themselves and the system would be entirely safe if laid on the surface of the ground.

22 The principal objection urged against the use of steel for this purpose is the liability to corrosion, especially when complicated by electrolysis in city streets. After a careful study of the problem, it was decided that the advantages of this material outweighed the objections, for the following reasons:

a If the entire pipe system had to be renewed every ten years, the steel would still be more reliable and hence more desirable than cast iron, as the latter is liable to fail by breakage without previous warning. When deterioration of the steel pipe does occur, it makes itself known by increase of leakage, which can be detected and repairs made gradually, as occasion offers. By way of a reductio ad absurdum it might be argued that if absence of corrosion be the controlling feature in the selection of a material, a glass pipe would be the ideal, as it would last forever if not broken by shock or bending stresses.

b The study of corrosion of steel has reached a point where it is possible to say that corrosion is not inevitable, but is due to more or less direct violation of certain well defined principles. If steel can be protected from the

simultaneous action of moisture, air and acids, the causes of rusting or corrosion are to a great degree removed, regardless of the particular chemical theory held by the investigator. Protecting the steel from the action of these three agents is mainly a mechanical proceeding. A permanent, impervious, elastic coating, which will adhere closely to the metal is the requirement. High-grade asphalt applied when both the steel and the coating are hot and clean, seems to meet these conditions satisfactorily. The specifications for the chemical composition of the asphalt used were very exacting. Among other conditions it was required that a cubic centimeter of the material should show no action when exposed for one year in any or all of the following solutions: 25 per cent hydrochloric acid, 25 per cent sulphuric acid, 25 per cent potassium cyanide, 25 per cent caustic soda, saturated solution of ammonia. The pipes were thoroughly cleaned and heated to a temperature of 300 deg. fahr. and while hot were dipped vertically in the bath of asphalt, which was maintained at a temperature of from 350 to 400 deg. fahr. The pipe was held in the bath for a sufficient length of time and was then drawn out slowly, at the rate of 5 to 10 ft. per min., so that a coating of 1/32 in. thickness was evenly distributed over the entire surface of the pipe. Any damage to the coating during shipment or erection was repaired by the application of the same material dissolved in a suitable solvent, and applied in several coats at intervals, until a satisfactory thickness was obtained. All bolts and nuts were dipped in the same solution before being inserted in the flanges.

c Electrolysis is due to an electrical difference of potential between the metal and the earth in contact with it, of sufficient magnitude to cause a current flow from the pipe to the earth. Since the conflagration of 1904 the City of Baltimore has required all electrical wires to be placed in the municipal conduits, and to protect the cables from electrolytic action the city's electrical commission made a careful study of the local situation. As a result the street railway was compelled to rebond a

large part of its tracks, copper cables being carried around all special work. In addition a supplementary copper return covering the entire district was installed, consisting of three bare copper cables having an aggregate cross-section of 6,000,000 circular mils. An entirely separate copper return was also installed as a protection for the cable sheaths, nothing but lead cables being bonded to this. The cross-section of this latter varied from 1,000,000 to 6,000,000 circular mils. As a result of this large amount of copper in the return circuit, the difference of potential between the various underground structures has been reduced to a nominal figure.

The special joint designed for the Baltimore steel pipe line has a resistance equal to 6 in. of the pipe on the 10-in. size, and equal to 9 in. on the 16-in. pipe. The resistance of a bell and spigot lead joint is often equal to 4 or 5 ft. of the pipe.

d Finally, the current flow from the pipe to the earth may be made very small if a high resistance covering be placed around the entire pipe system. In the present instance the asphalt coating furnishes the necessary high resistance envelope.

23 While it is still too early to offer definite evidence regarding the life of the pipe in question, pipes which have been in the ground for two years have recently been exposed by excavation for other work, and have shown absolutely no signs of deterioration, the coating being in perfect condition.

24 *Pipe.* The specifications called for lap-welded pipe made of soft open hearth steel having the following qualities:

	Per Cent
Carbon, not exceeding	0.10
Phosphorous, not exceeding	0.04
Sulphur, not exceeding	0.05
Manganese between	0.35–0.45

Ultimate tensile strength between 50,000 and 55,000 lb. per sq. in.; elastic limit at least ½ ultimate; elongation not less than 20 per cent in 8 in.; cold and quench bend 180 deg. flat.

25 The weld in the lap was to be perfect and capable of standing the strains incident to the manufacture of bends and forming of joints, without distress or rupture.

26 The thickness of pipe, in inches, was as follows:

24 in. Outside Diameter	$\frac{1}{2}$
16 in. Outside Diameter	$\frac{7}{16}$
10 in. Inside Diameter	$\frac{7}{16}$
8 in. Inside Diameter	$\frac{7}{16}$

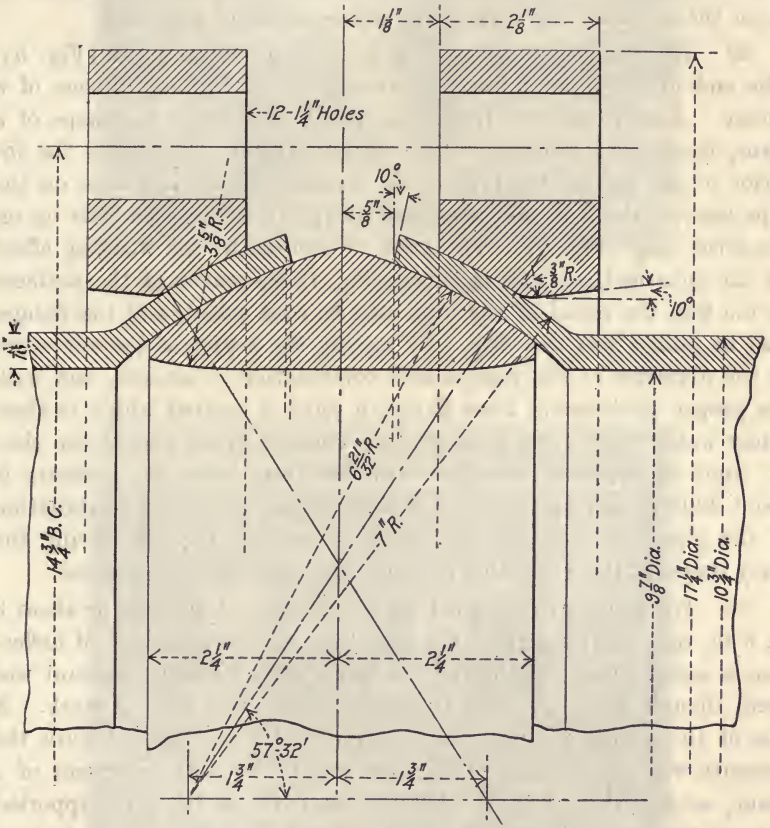


FIG. 5 DETAIL OF 10-IN. PIPE JOINT

27 Bends were generally of a standard radius of 5 diameters of pipe. Length of pipe, in feet, laid was:

24 in	1,275
16 in	17,052
10 in	28,229
8 in	7,137
Total	53,693 = 10.2 miles

28 *Joints.* For the present service it was desired to avoid the use of rubber gaskets, which would tend to increase the electrical resistance of the joint. Copper gaskets or other metals dissimilar to the metal of the pipes, in the presence of moisture and the acids or salts of the earth, would form a voltaic cell, and tend to increase the corrosion due to electrical action. A joint was therefore designed to avoid the necessity for a gasket or joint cement of any kind.

29 The Baltimore design is a form of universal joint (Fig. 5). The end of the pipe is flanged out into a bell forming a zone of a sphere. A soft cast-iron ring is accurately turned in the shape of a torus, having the same curvature on its exterior surfaces as the interior of the bell on the pipe ends. Loose flanges are placed on the pipe back of the bell, and when bolted up, draw the pipe bells up on the torus ring (Fig. 6). The pressure secured by the wedging effect on the spherical surfaces is enormous. If the curve of the surfaces be too flat, the metal of the pipe may be cold rolled, and the flanges may be pulled over the ring. By suiting the degree of the curvature to the diameter of the pipe, a safe combination is secured, and with the proper thickness of loose flange, a joint is secured which is absolutely water tight until pressures are reached which exceed the elastic limit of the pipe or bolts. On the 10-in. size this pressure is about 2200 lb. per sq. in. In the field testing, during the installation of the pipe, the specified test pressure was 600 lb., but during the early stages of the work, this pressure was often largely exceeded.

30 The joints are designed for a deflection of 10 deg., or about 3 ft. 6 in. in a 20-ft. length. On the 10-in. size this amount of deflection is easily obtainable, but on the larger sizes a smaller amount was used, though sufficient to be of considerable value in city work. A line of 10-in. pipe was made up of seven 20-ft. lengths, and while the pressure was on, one end of the line was raised 5 ft. by means of a crane, as shown in Fig. 7. While suspended in the air, supported only at the two ends, 140 ft. apart, there was practically no leakage at any of the joints. It is evident that at least that amount of trench might be washed out without interfering with the operation of the pipe system in the least degree. In laying the pipe it was possible to deflect it to pass an obstruction and in one instance, after the completion of the work, owing to a change of grade in the sidewalk, a hydrant branch 50 ft. long was jacked up 5 in. at one end, simply by loosening the bolts at one joint. Upon tightening these bolts, the line tested free of leaks at 600 lb.

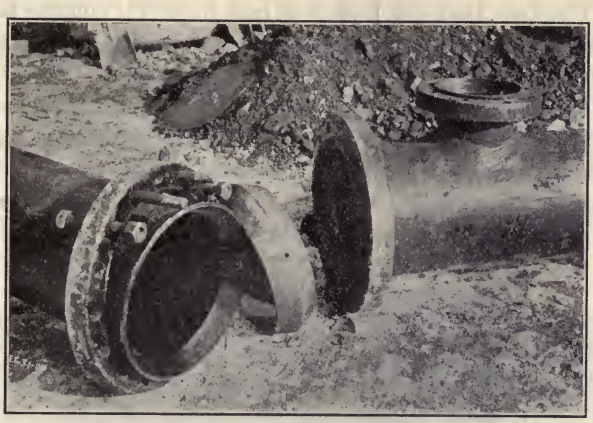


FIG. 6 VIEW OF PIPE JOINT AND WELDED NECK

FIG. 7 SHOP TEST OF PIPE JOINT

31 The fact that the pipe was laid in 20-ft. lengths means that the number of joints was reduced 40 per cent in comparison with cast iron.

32 For deflections greater than could conveniently be made by the joints, pipe bends were used. A bending table was installed on the work by the contractor, and about 1500 bends were made, or 150 per mile of pipe laid. Bends of 16-in. pipe and smaller were made

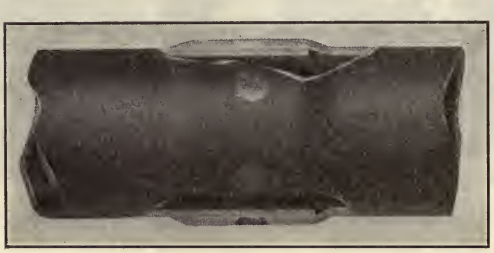


FIG. 8 DETAIL OF STRAIGHT LINE WELD

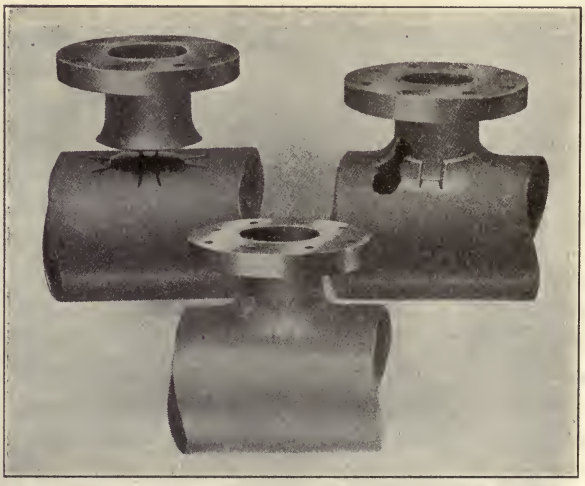


FIG. 9 DETAIL OF WELDED NECK

on the work, the 24-in. bends were made at the factory before shipment.

33 For make-up pieces at intersections and in blocks between valves or tees already installed, straight line welded joints were used. A special joint was devised for this service, made up as follows: The end of one pipe was accurately expanded sufficiently to

permit of its being shrunk over the end of the pipe to which it was to be joined. Holes were cut around the circumference of the outer pipe or bell, and after being heated it was shrunk on in place. The holes were then flowed up with metal by the oxy-acetylene blow-pipe, and the end of the bell was also welded to the enclosed pipe. With this type of joint the weld is in shear and not in tension, and it is entirely feasible to make bends in the pipe with the weld in the arc of the curve. Approximately 1500 of these straight line welds were made on the work, or about 150 per mile of pipe (Fig. 8).

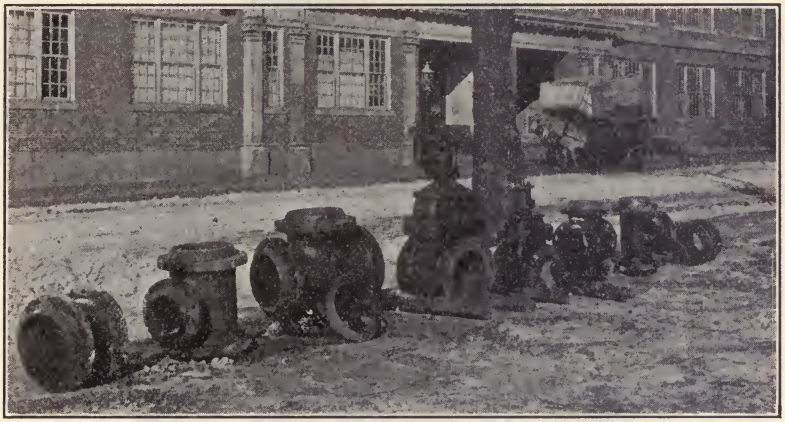


FIG. 10 CAST STEEL SPECIALS

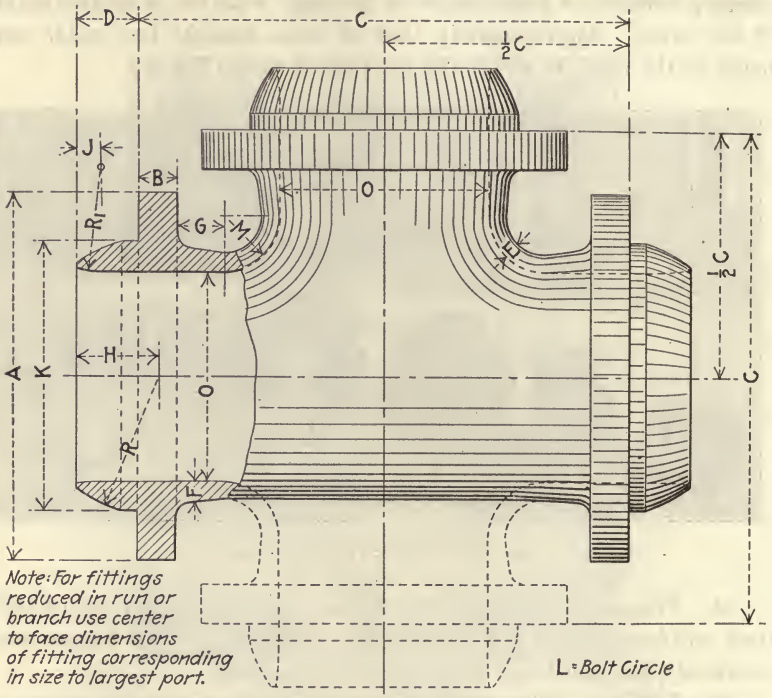
34 *Flanges.* The loose ring flanges were of medium open hearth steel, ultimate tensile strength 60,000 to 68,000 lb. All flanges were accurately machined and were drilled to template.

35 *Fittings.* The use of steel pipe made it possible to reduce the number of fittings to a minimum. The hydrant branches were made by welding necks to the mains and laterals. A special neck weld was used, made up as follows: A hole was cut in the pipe, smaller than the size of the neck, and radial cuts were made forming four narrow lugs which were left projecting into the hole. The wider alternate lugs were bent back to make an opening large enough to receive the neck piece. The smaller straight lugs formed a support for, and held the neck rigidly in place during the welding process. The whole joint was then flowed with metal by an oxy-acetylene blow-pipe, forming a joint as strong as the original pipes (Fig. 9).

36 The only specials required were the tees and crosses at the intersections of the gridiron, and a valve connection piece used to

give a straight face flange at the valves. By the use of the latter it was possible to use standard commercial valves, and the valves may be easily removed when necessary (Figs. 10, 11, 12). All fittings were made of low carbon open hearth cast steel.

37 *Valves* All valves are double-disk parallel seat gates. Bodies are low carbon open hearth steel of the same specifications as for

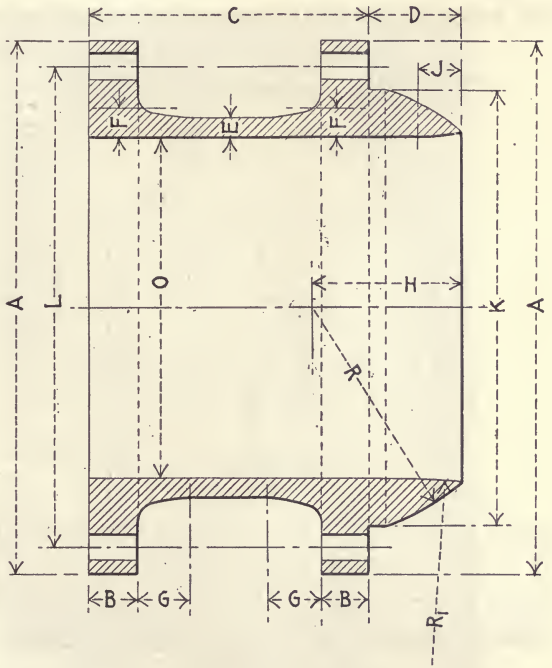


SIZE	A	B	C	D	E	F	G	H	J	K	O	R	R_1	L	M	BOLTS NO.	BOLTS DIA.	HOLE DIA.
10"	17¼"	2⅛"	23½"	3"	13/16"	1⅛"	2¼"	4"	1⅛"	12 27/32"	10"	6 21/32"	3 5/8"	14¾"	2⅝"	12	1⅛"	1¼"
16"	23½"	2½"	30"	4⅜"	⅞"	1¼"	2½"	6"	½"	20 13/32"	15⅛"	9 29/32"	6¼"	21"	2¾"	16	1⅛"	1¼"
24"	33"	2¾"	41"	3⅞"	1 1/16"	1½"	3"	8⅜"	2¼"	27"	23"	14 19/32"	7"	30"	3"	24	1¼"	1⅜"

FIG. 11 DETAIL OF TEES AND CROSSES

fittings. Bonnets, disks, gearing brackets, glands and packing boxes are of semi-steel. Stems are of Tobin bronze, not less than 53,000 lb. tensile strength. Seats, wedge mechanism and glands are of bronze. The 16-in. and 18-in. valves are geared. All street valves are provided with a stem nut, and a forged steel key is placed in each valve box. An interlocking arrangement is provided so that the key

can be removed from the nut only when the valve is wide open. This prevents the valves being left closed or opened only a few turns, except in an emergency or intentionally, as the valve box cover cannot be replaced while the key is on the nut. The 24-in. valves are in the pumping station and are hydraulically operated. All valves were subjected to a shop test of 800 lb. when open, and 600 lb. when closed.



SIZE	A	B	C	D	E	F	G	H	J	K	R	R_1	O	L	BOLTS		
															NO.	DIA.	HOLE DIA.
8"	$14\frac{3}{4}$"	$1\frac{7}{8}$"	$10\frac{1}{2}$"	$2\frac{15}{16}$"	$\frac{3}{4}$"	$1\frac{1}{8}$"	$2\frac{1}{4}$"	$3\frac{1}{4}$"	1"	$10\frac{9}{16}$"	$5\frac{1}{4}$"	$3\frac{1}{2}$"	$7\frac{3}{4}$"	$12\frac{1}{4}$"	8	$1\frac{1}{8}$"	$1\frac{1}{4}$"
10"	$17\frac{1}{4}$"	$2\frac{1}{8}$"	$11\frac{1}{2}$"	3"	$\frac{3}{4}$"	$1\frac{1}{8}$"	$2\frac{1}{4}$"	4"	$1\frac{1}{8}$"	$12\frac{27}{32}$"	$6\frac{21}{32}$"	$3\frac{5}{8}$"	$9\frac{7}{8}$"	$14\frac{3}{4}$"	12	$1\frac{1}{8}$"	$1\frac{1}{4}$"
16"	$23\frac{1}{2}$"	$2\frac{1}{2}$"	13"	$4\frac{3}{8}$"	$\frac{7}{8}$"	$1\frac{1}{4}$"	$2\frac{1}{2}$"	6"	$1\frac{1}{2}$"	$20\frac{13}{32}$"	$9\frac{29}{32}$"	$6\frac{1}{4}$"	16"	21"	16	$1\frac{1}{8}$"	$1\frac{1}{4}$"
24"	33"	$2\frac{3}{4}$"	$14\frac{1}{2}$"	$3\frac{7}{8}$"	$1\frac{1}{16}$"	$1\frac{1}{2}$"	3"	$8\frac{3}{8}$"	$2\frac{1}{4}$"	27"	$14\frac{19}{32}$"	7"	23"	30"	24	$1\frac{1}{4}$"	$1\frac{3}{8}$"

FIG. 12 DETAIL OF VALVE CONNECTING PIECE

38 *Valve Boxes.* The street valves are treated as pieces of mechanism, subject to derangement, and hence are placed in concrete boxes or manholes, 42 in. inside diameter, where they can be inspected and repaired without disturbing the street paving. The

floor was first laid of plain concrete, with suitable drainage connection; then the section enveloping the pipe or fitting was laid up with concrete blocks molded to the standard radius. The upper clear section was built of reinforced-concrete rings, laid up without mortar, as they are provided with interlocking projections on the top and bottom faces, which make the rings self-centering. The upper portion is made of conical shaped rings, forming the roof, and supporting the cast-iron cover frame. The blocks and rings were made up in quantities in metal molds and seasoned before being used. They

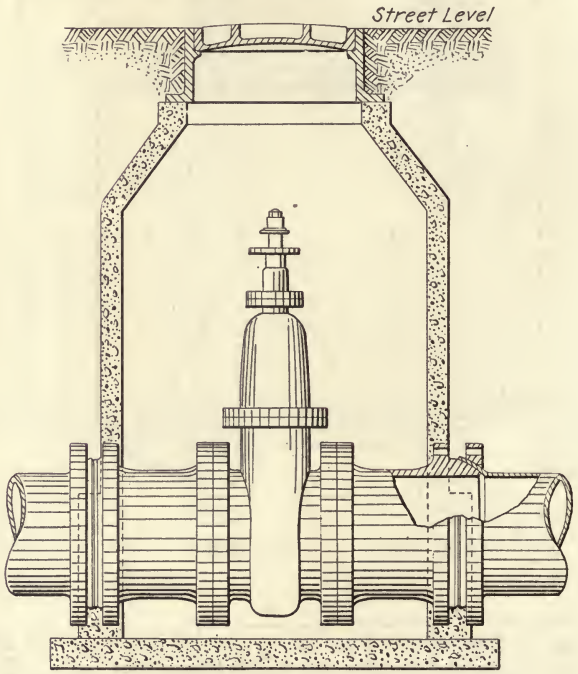


FIG. 13 VALVE BOX WITH VALVE IN PLACE

were quickly and easily assembled in the trench by unskilled labor. The holes were back-filled and the street repaved in a remarkably short time, an item of considerable importance in a business thoroughfare as there was no time required for setting of the concrete (see Figs. 13 and 14).

LOW-PRESSURE WATER SUPPLY

39 Two separate 40-in. mains from two different river systems, run direct to and from the normal supply for the high-pressure pump-

ing station. The normal pressure available with the high-pressure draft included is from 30 to 50 lb., but if by-passed from the middle or high-service mains, the pressure can be carried at 70 lb. A 30-in. cast-iron branch is taken to the station where it changes to a 30-in. rivetted steel pipe inside the building.

40 An auxiliary supply of brackish water from the harbor may be secured through a reinforced-concrete conduit 3 ft. sq. inside.

FIG. 14 CONCRETE VALVE BOX MADE UP

The conduit is provided with a large screen chamber at the dock, and has fine and coarse screens sliding in inclined frames, and both in duplicate, so that one set of each can be in service while the other set is out for cleaning. The screen chamber is also provided with stop logs so that the chamber and conduit can be pumped dry for cleaning out sediment. At the station end the conduit terminates in a large suction chamber located beneath the basement floor and between the pump foundations. The dimensions of this chamber

are 8 ft. wide, 7 ft. deep and 70 ft. long. The gravity suction pipes enter the suction chamber through cast-iron thimbles built into the floor of the basement, as the basement floor is 8 or 9 ft. below flood tide level.

41 A by-pass from the high-pressure discharge main is led into the suction chamber for flushing it, and also forms a convenient discharge for the pumps during tests.

INSTALLATION OF PIPE LINES

42 *Tunnels.* A considerable amount of tunneling was done, especially at street intersections, where double track crossings of the street railway were frequently encountered. On several of the busiest streets it was found necessary to tunnel for many blocks at a stretch. In a large number of instances it was thought desirable to tunnel for the hydrant branches, as the hydrants being staggered on opposite sides of the street, it was necessary on every alternate hydrant to cross the street, thus crossing also the railway tracks, pipes, sewers, electric conduits, etc. A segregation of these tunnels into groups is as follows:

	Lineal Feet
Intersections	5,122
Straight line	4,623
Hydrant connections	2,630
Total tunnels	12,375

43 *Leakage.* For field tests during the installation of the pipes, the contractor furnished a portable testing set consisting of a 3-h.p. 4-cycle gas engine, driving through a Morse chain a triplex plunger pump of a capacity of 5 gal. per min. at a pressure of 1000 lb. per sq. in. Each street section between valves was tested as the work was installed, and made absolutely tight, at the specified pressure of 600 lb: The section to be tested was filled from the ordinary fire hydrants, after which the test pump was connected through a 1-in. flexible steel hose. In the early stages of the construction, the workmen would frequently carry the pressure up to 800 lb. and sometime to 100 lb., to show interested spectators what the new joint would stand. Afterwards a relief valve set for about 610 lb. was placed on the pump, and the testing became almost automatic.

44 After the pipe lines had been completed, but before the system had been put into commission, a pressure of about 300 lb. was put on the pipe lines by the main pumps. There being no hydrants open,

the water was being wasted through a by-pass to reduce the pressure. A suggestion was made to close the pipe system off from the pumps to determine how long the pressure would remain. The main 24-in. valves were accordingly closed, while the pressure was at 125 lb. After a lapse of 16 hours the gage registered 95 lb., indicating that the leakage was practically nothing.

45 A duplex pump of 100 gal. capacity was installed to keep up the pressure and take care of the leakage on the system. Since the equipment has been put into regular service it has been necessary to keep the by-pass valves open continually on this pump in order to permit the plunger to move.

CHOICE OF MOTIVE POWER

46 *General Considerations.* In the preliminary studies for the project, careful attention was given to the design of similar plants in other cities, notably New York and Philadelphia. In the former, as is well known, electric driven centrifugal pumps are used. In Philadelphia, gas engine driven, geared triplex plunger pumps were installed. In the following discussion it should be remembered that local conditions often exercise a controlling influence in the decision as to the best form of motive power for such a plant.

47 In Baltimore the fundamental principle governing the design of the entire project was considered to be reliability under the most adverse conditions liable to be encountered. When it is considered that the safety of hundreds of millions of dollars worth of property is dependent upon the accurate functioning of each element, it is evident that no considerations of economy, either in first cost or operating expense, should be allowed to enter, if simplicity and certainty of operation are thereby subordinated.

48 *Gas Engines.* The gas engine is subject to certain limitations, which in the case of fire service, place it at a distinct disadvantage.

a The load factor of the service is extremely low, being less than 5 per cent per annum, so that the opportunity to profit by the high fuel economy is negligible.

b Inability to carry more than a small percentage of overload.

c In the hands of expert operators, a well designed modern gas engine is perhaps subject to no more accidents or delays than is a steam equipment, but certainly it is not subject to any *less* than steam. On the contrary, how-

ever, in the hands of operators of only average intelligence or experience, the number of apparently trivial causes which can result in serious delays or damage is surprisingly large in a gas engine plant. In the case of a municipal plant of this type it is useless to plan for ideal operating conditions, or to assume that only the highest grade operating force will be employed.

49 *Electric Motors.* In spite of its many known advantages, a large electrical distributing system is essentially in unstable equilibrium, and subject to complete interruption from very slight causes, either natural or malicious. Dependence for prompt renewal of the service after a shutdown must be placed in duplicate lines and equipment in the stations. But even with these a considerable period must elapse before large electrical machines can be started up and brought into synchronism. A delay of this character during the first critical minutes of a bad fire would be fatal to the usefulness of an important fire fighting system. The fact that a vital feature of the fire fighting system would be under the control of employes of an outside corporation and not subject to the discipline of the fire department, with the possibility of a conflict of authority at a crucial moment, are all matters that must be considered.

50 Like the gas engine, the electric transmission, especially the underground system, requires for its commercially profitable conditions a high load factor, but for a different reason. To justify the large investment needed, requires a uniform load as near as possible to its capacity, in order that the fixed charges shall not be out of proportion to the earnings. Owing to the extremely low load factor of a fire station, its load is a very "undesirable" one to a commercial electrical supply corporation, unless a "demand" charge is made sufficient to justify holding in reserve a definite proportion of the entire equipment from the coal pile to the electric cables. Electrical plants require the installation of sufficient equipment all along the line to supply the maximum annual peaks. As a fire is no respector of anybody's peaks, it follows that a fire station load demands its own separate power plant investment, whether the equipment is located in a commercial central station, or in an isolated plant for its own use.

51 A commercial plant is designed with a definite load factor in view, ranging usually from 30 to 50 per cent. For these conditions the most efficient equipment is justified, with all the refinements of

modern fuel and labor saving auxiliaries in a large plant. If a load factor of only 5 per cent had been imposed, however, a very different type of equipment would have been selected, at a very much smaller investment.

52 The various links in the chain of an electrical supply equipment (not considering a long distance transmission) would be as follows:

a Coal handling apparatus
b Boilers and auxiliaries (stokers, stacks, economizers, heaters, etc.)
c Steam turbines and auxiliaries
d High-tension generators
e High-tension switching apparatus
f High-tension cables

and at the receiving end

g High-tension switches
h High-tension motors
i Centrifugal pumps

53 If a long distance transmission were included, there would be added step-up transformers, aerial lines, a substation with step-down transformers and switching apparatus. If low-tension motors were used, additional transformers would be required.

54 An isolated steam pumping station, designed for the purpose, eliminates at once five of the above links, namely, items *d, e, f, g* and *h,* and concentrates the entire operation in one building, under the direct control of the fire fighters themselves.

55 *Finances of Steam and Electrical Operation.* Expressed in dollars and cents the argument becomes as follows:

Electric Pumps (New York Type)		
Investment		
5 motor-driven pumps (rated capacity 3000 gal. per min.), switchboard, etc	$112,500	
Building and pump foundations	84,000	
		$196,500
Operation		
Maintaining pressure continually 8760 hr. less 100 hr. = 8660 hr. at 100 kw	866,000 kw-hr.	
Fire service, 100 hr. per annum 3150 kw. demand	315,000 kw-hr.	
		1,181,000 kw-hr.

Service charge, maximum demand =	3150 kw.	
Central station investment, 3150 kw. at $75	$236,000	
Underground cable (Baltimore conditions)	40,000	
Cash requirements	276,000	
Underwriting at 90	31,000	
Total investment		$307,000
Fixed charges on $307,000		
Interest	at 5 per cent	
Depreciation	at 5 per cent	
Profit	at 5 per cent	
Total	15 per cent	$46,000
Underground conduits, duct rental (Baltimore conditions)	1,300	
Total service charge		$47,300
Operating expenses		
Service charge	$43,700	
Meter charge, 1,181,000 kw-hr. at 1 cent	11,810	
Salaries, station operating force	10,650	
Supplies, lubrication and repairs	1,000	
		$67,160
Fixed charges on $196,500		
Interest at 4 per cent	$7,860	
Depreciation at 5 per cent	9,825	
		17,685
Total annual expense, electrical plant		$84,845
STEAM PUMPS		
Investment		
Four 4000-gal. pumps and auxiliaries	$86,000	
Boilers and auxiliaries	70,000	
Piping, steam and auxiliary water	30,000	
	$186,000	
Building and machinery foundations	125,000	
		$311,000
Operation		
Coal consumption		
Banking fires, 8760 hr.—		
100 hr. = 8660 hr. =		
360 days at 6 tons per day	2160 tons	
Fire service, 100 hr. per annum at 5 tons coal per hour	500 tons	
Total	2660 tons	

Operating expenses		
Coal, 2660 tons at $3.30	$8,778	
Salaries, station operating force	13,350	
Supplies, lubrication and repairs	2,000	
		$24,128
Fixed charges on $311,000		
Interest at 4 per cent	$12,440	
Depreciation at 5 per cent	15,550	
		27,990
		$52,118

SUMMARY

Total annual expense, electrical plant	$84,845
Total annual expense, steam plant	52,118
Total annual saving	$32,727

This saving capitalized at 9 per cent represents an investment by the city of $363,630, considerably more than the first cost of the steam plant in the above comparison.

STEAM PUMPING STATION

56 *Pumps.* The Baltimore plant is designed for four main units of 4000 gal. per min. rated capacity, at a piston speed of 300 ft. per min. and making 50 r.p.m. Three main engines have been installed and are in operation at present; the fourth unit will be added in the near future (Fig. 15).

57 In line with the policy of designing all parts of the system with a first requisite of simplicity and reliability, the main units each consist of a horizontal, twin, simple, non-condensing, crank and flywheel, plunger pumping engine. The water ends are attached directly to the engine frames, at opposite ends from the steam cylinders, the crankshaft being in the center (Fig. 16).

58 The steam cylinders are fitted with standard Corliss valve gears, having double eccentric long range cut-offs. The cut-offs of both cylinders are under the direct control of the speed and pressure regulators, and are also provided with hand control.

59 Each engine was liberally designed for a continuous working pressure of 300 lb. per sq. in. on the water ends, with a test pressure of 600 lb. static. Injection parts were designed for a variation of pressure from 70 lb. direct to a suction lift of 15 ft. of salt water. All steam parts were designed for maximum working pressure of 200 lb., but the normal working pressure is only 125 lb.

60 Each engine is fitted with speed and pressure governors. The speed governor is driven by a noiseless chain belt, and acts directly on the cut-off valves of both cylinders. The pressure governors are identical in principle with the regulating valves used on the hydrants, and were also furnished by the Ross Valve Manufacturing Company.

61 The net effective valve area between the openings of the valve seats on each suction and discharge deck is 308 sq. in. or 220 per cent of the cross sectional area of the plungers. The valves are 3½ in. in diameter and are composed of rubber with brass backing plates; the seats are of bronze, screwed into the valve decks on a taper and faced off after being placed in the decks.

62 *Auxiliaries.* To take care of the leakage in the pipe system and maintain a pressure of 150 lb., as well as to provide for the first draft from the hydrants before the main pumps are in action, a 1000-gal. per min. pump was installed. This is a horizontal, duplex, direct-acting, compound, non-condensing, center-packed plunger pump, giving its rated capacity at a piston speed of 100 ft. per min. While the pump was designed for a normal working pressure of 150 lb. on the water end, and for 125 lb. on the steam end, all pressure parts were designed to withstand a continuous pressure of 300 lb. on the water end and 200 lb. on the steam end, so that the pump could be left in service under the maximum pressures of the main pumps without injury. Owing to the leakage on the system being so small, the pump is inconveniently large for the purpose intended, and it has been necessary to keep the delivery by-passed continuously in order to keep the plunger in motion.

63 In order to maintain the air in the delivery air chambers, there are provided two steam driven, crank and flywheel, two-stage air compressors, each having a capacity of 50 cu. ft. of free air per min. against a pressure of 450 lb. There is also provided a wrought steel storage tank 30 in. in diameter by 5 ft. high. There are two large air chambers on the 24-in. discharge mains, in addition to those on the pumps. The former are made of lap-welded rolled steel pipe, ½ in. thick, 30 in. in diameter and 20 ft. high. The ends are bumped, riveted and welded to the pipe. Two sets of glass gages are attached through bosses welded on the sides.

64 For priming purposes, when lifting from the harbor, there is provided one vertical, steam driven, crank and flywheel single-stage, dry vacuum pump, 6 in. steam, 10 in. air, and 6 in. stroke. This pump is connected to the harbor suctions through a large separator

FIG. 15 INTERIOR OF STATION

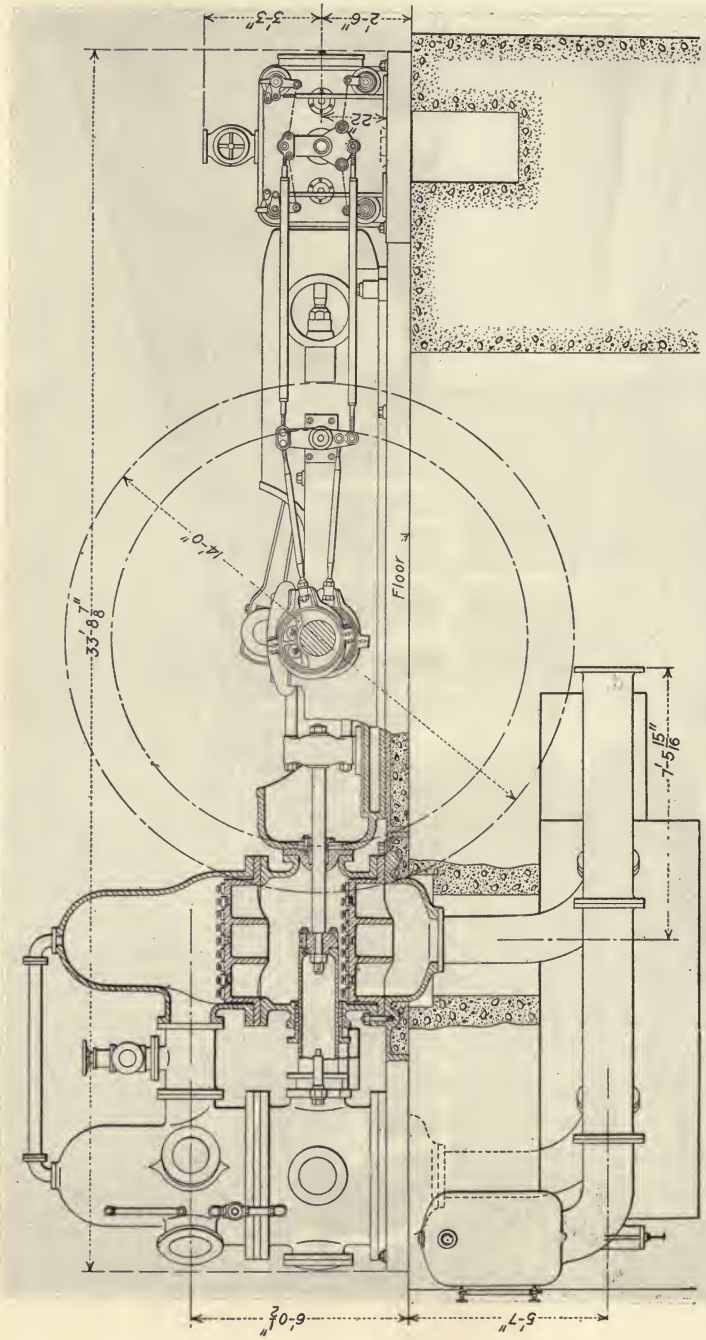


FIG. 16 ELEVATION OF PUMP

tank at the top of a 40 ft. riser, to prevent water being drawn over into the air cylinder.

65 The plant is provided with a four-motor, electric traveling crane, main hoist 20 tons capacity, auxiliary hoist 5 tons. For lighting the station and for the operation of the electrically driven auxiliaries, there are provided two non-condensing steam turbo-generators.

66 *Tests and Duty Trials.* The specifications for the pumps provided for an endurance and capacity test of 24 consecutive hours.

67 A normal load duty trial was also specified, covering a period of 12 hours; steam pressure at the throttle 125 lb. The steam consumption was to be based on the feedwater supplied to a separate boiler blanked off from all other sources of supply. The measured consumption was to include all jacket steam, but not that required for boiler feed pumps and other auxiliaries, nor separator condensation and drips from the steam piping on the boiler side of throttle.

68 In order to avoid the insertion of a water meter in either the main suction or discharge lines, a venturi meter is by-passed around a gate valve in the 30-in. city water supply. As the pump discharge can be accurately measured from the record of the pump strokes for a given period, when the slip is known, in this case the venturi meter is only used to calibrate the slip of a single pump, and therefore has a capacity only equal to one pump.

69 The duty specified is 70,000,000 ft.-lb. per 1000 lb. of dry steam, under normal operating conditions of 300 gal. per min. discharge, against a head of 250 lb., with 30 lb. pressure on the suction, and 125 lb. of steam at the throttle.

70 *Boilers.* The plant is designed for four boilers, each set singly, and each provided with a separate stack carried on structural steel supports directly over the setting. At this time only three boilers have been installed, the fourth will be added in the near future (Fig. 17).

71 The boilers are of the horizontal, inclined, straight tube type, with forged steel water legs reinforced with hollow staybolts. Each boiler contains 6800 sq. ft. of heating surface, and three 36-in. drums. All pressure parts are designed for 200 lb. working pressure. Each boiler is capable of a continuous evaporation of 45,000 lb. of water from and at 212 deg. fahr. into commercially dry steam, when burning semi-bituminous coal of approximately 14,500 B.t.u., with forced

draft not exceeding 3 in. of water in the ashpit, or the equivalent of 1 boiler h.p. from 3.77 sq. ft. of heating surface.

72 As the actual time during which the boilers will be in active service will probably average only about 100 hours per annum, the plant was designed for carrying banked fires for a large proportion of the time, with the least possible loss from radiation. The top, sides and rear of each boiler are enclosed in an air-tight steel plate casing. The hotter portions of the side walls, amounting to about 20 per cent of the area of these walls, are covered with 2 in. of magnesia blocks inside the casing. The casing plates are supported on a framework of steel angles, attached to the structural steel members supporting the boilers and stacks. These angles act also as buckstaves for the settings. Access and dusting doors are provided where necessary, which are hinged at the top and close tightly on inclined faces.

73 The boilers are set in the reverse direction from the usual method, that is, with the low end of the tubes over the furnace. The front portion of the furnace is covered with a flat fire brick arch, made of split tiles encircling and supported by the lower row of tubes. When under fire, there is presented to the furnace gases an incandescent fire brick surface, instead of the customary cool iron surface of the tubes. The first pass of the gases between the tubes is at the extreme rear of the boiler, thus providing a furnace area equal to the entire floor space occupied by the boiler inside the setting, and making it possible to force the boiler to 90 per cent over the customary rating without imperfect combustion. A special grade of fire brick was used for the furnace lining, which under a change of temperature of 3100 deg. fahr. shows an expansion or contraction of less than 0.01 in. per ft.

74 *Forced Firing.* The conditions of fire service require that the boilers shall be capable of changing from banked fires to the maximum capacity in the shortest possible time. The intervals specified were as follows: one-half rated capacity in 5 minutes; full rated capacity in 12 minutes; overload of 75 per cent in 20 minutes.

75 *Choice of Fuel.* The above conditions made imperative the use of a gaseous fuel, or fuel oil, or forced draft with coal. The possibility of the use of a combination of two of these was also considered. With gas at 90 cents per 1000 ft., this material was abandoned. The use of crude oil was given careful study. Due to the fact

that economy of operating expense was not the primary consideration, the problem was somewhat simplified.

76 As the plant is located in a wholesale district where high values of stock are common, it would have been necessary to have stored the main supply of oil underground. Ample space for this purpose existed in the wide water front street about 200 ft. from the station. On the other hand, experience has demonstrated that oil should be fed to the burners only by gravity from an overhead storage of suitable capacity. Experience has also shown that in spite of all precautions, an oil storage tank will in all probability, sooner or later, take fire. It was, of course, possible to design a fireproof barrier which would prevent the fire doing damage to adjoining property. It was believed, however, that in a plant designed for fighting conflagrations, the possibility of an oil fire, with its huge volumes of dense black smoke, would not tend to popularize the system, even if the fire proved to be entirely harmless. In spite of its recognized advantages in ease of handling and quick firing, the use of fuel oil was therefore definitely abandoned, and coal was adopted. The introduction of natural gas from the West Virginia fields has been under consideration for some years, and if this should be accomplished, it would make an admirable fuel for the purpose, either alone or as supplementary to the coal furnaces.

77 *Automatic Mechanical Stokers.* The use of coal made necessary also the use of forced draft. In addition, it was desirable that during the periods of banked fires, there should be maintained a full bed of ignited coal, requiring only the air blast to force the fire to the highest rate of combustion. The underfeed type of stoker seemed to meet these conditions very satisfactorily. A large body of coal, approximately 1000 lb., can be carried in each furnace, a part of this coal incandescent, a part coked and a part in the process of coking. This represents about 14,500,000 B.t.u. in storage ready for use on a few minutes' notice. As a further heat storage, the steam pressure, which is ordinarily carried at about 150 lb., can be raised to 200 lb., immediately upon the receipt of an alarm. As the normal operating pressure is only from 125 to 150 lb. at the pump throttles, by the time the hose companies can reach the hydrants and attach the hose, and put a sufficient draft on the pumps to pull the steam down to normal, the furnace fires will be ready to respond to any demand.

78 Each boiler is equipped with four underfeed stokers, and each stoker unit is capable of burning efficiently, without smoke, 1000 lb.

of coal per hour. For the four boilers a duplicate blower equipment is installed, each consisting of a full housed steel plate fan of sufficient capacity to operate the four boilers at 75 per cent overload. Each blower is driven by a direct-coupled vertical steam engine, and is connected to the main air pipe line, and by means of dampers either blower can be used to serve any or all the boilers.

79 To regulate the supply of both fuel and air in the proportion required for complete combustion under all rates of firing, the stokers are provided with automatic regulators actuated by the boiler pressure. Provision is also made for hand regulation, so that it is possible to anticipate sudden demands for steam upon receipt of an alarm, and also for decreasing the supply of both fuel and air when the demand for forced firing has passed.

80 *Stacks.* Each boiler is provided with a separate steel stack located directly over the boiler which it serves. Each stack is 72 in. diameter by 125 ft. above the boiler room floor, and is carried by structural steel framing from the boiler foundations. The stacks are unlined.

81 *Coal Handling and Storage.* Semi-bituminous run of mine coal is used almost entirely for steaming purposes in the vicinity of Baltimore. In this instance the coal is delivered in carts, the total annual consumption being too small to justify mechanical handling under existing conditions. The coal is dumped on a grating over an opening in the sidewalk, the mesh being 4 in. sq. Large lumps are broken up with a maul, as with the type of stoker in use there is no necessity for the coal to be crushed to smaller sizes. Below the grating there is a dumping hopper which receives the coal and delivers it to the lower run of a bucket elevator. The elevator is a double strand link belt with V-shaped buckets, 16 in. by 15 in., dumping by gravity. The outfit has a capacity of 25 tons per hour with uniform feed.

82 The coal is delivered into reinforced-concrete bunkers with inclined bottoms, located directly over the fireroom. The bunkers have a capacity of 150 tons without trimming, or sufficient to operate the entire plant at its maximum capacity for 30 hours. Coal can also be delivered from carts directly on to the boiler room floor, so that the operation of the plant is not dependent upon the coal handling equipment nor the storage supply.

83 *Boiler Feedwater.* Boiler feedwater is normally supplied from the city mains under 30 to 50 lb. pressure, sufficient to deliver

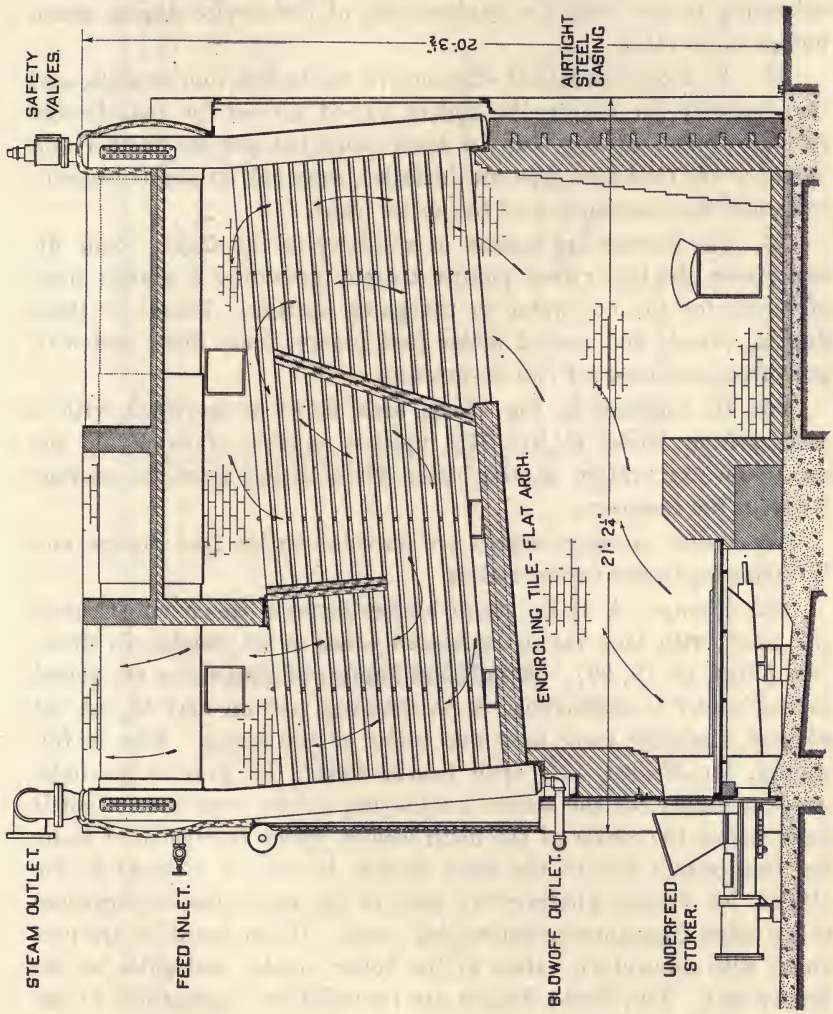


FIG. 17 SECTION OF BOILER

to the open heaters in the gallery of the engine room, without pumping. In the event of an interruption to the city supply, the feedwater can be taken from a reserve supply stored in steel tanks in the basement under the boilers. For lifting from the storage tanks and delivering to the heaters a duplicate set of low-service duplex steam pumps is provided.

84 To avoid the loss of efficiency in the boilers due to scale, and the necessity for taking the boilers out of service for considerable periods while removing scale, a double unit hot process purifier and heater of the Cochrane type was installed, each half of ample capacity to handle the consumption of the entire plant.

85 The heaters are located in a gallery in the engine room, directly over the boiler feed pumps, thereby providing a gravity head of 25 ft. for the hot water to the pump suction. There are three duplex, outside end packed boiler feed pumps, brass fitted and with pot valves, designed for 300 lb. pressure.

86 In addition to the above, each boiler is provided with a Metropolitan Model O No. 7½ injector, capable of supplying the maximum evaporation of the boiler when lifting from the storage tanks in the basement.

87 Foster excess governors are provided on the feed pumps, and Williams regulators on the boilers.

88 *Piping.* A 12-in. steam header forms a closed ring around the plant, with long radius expansion bends at all changes in direction (Figs. 18, 19, 20). A sufficient number of gate valves are placed in the header to sectionalize it, so that any portion may be cut out without disabling more than one boiler or one pump. Pipe is full weight, lap welded, soft open hearth steel. To provide an independent header for the station auxiliaries, a 6-in. cross connection is made across the center of the main header, which is capable of being fed from either side of the main header, in case of accident to the other. No fittings whatever are used in the main line, all branches being taken from interlocked welded necks. Boiler branches are provided with non-return valves at the boiler nozzles and gates at the header end. Van Stone flanges are provided for connections to the valves and receivers, which are located so as to avoid as far as possible the necessity for any additional joints in the line. Wrought steel receiver type separators are installed at the low points on each side of the header.

89 The exhaust system is extremely simple, a multi-port back

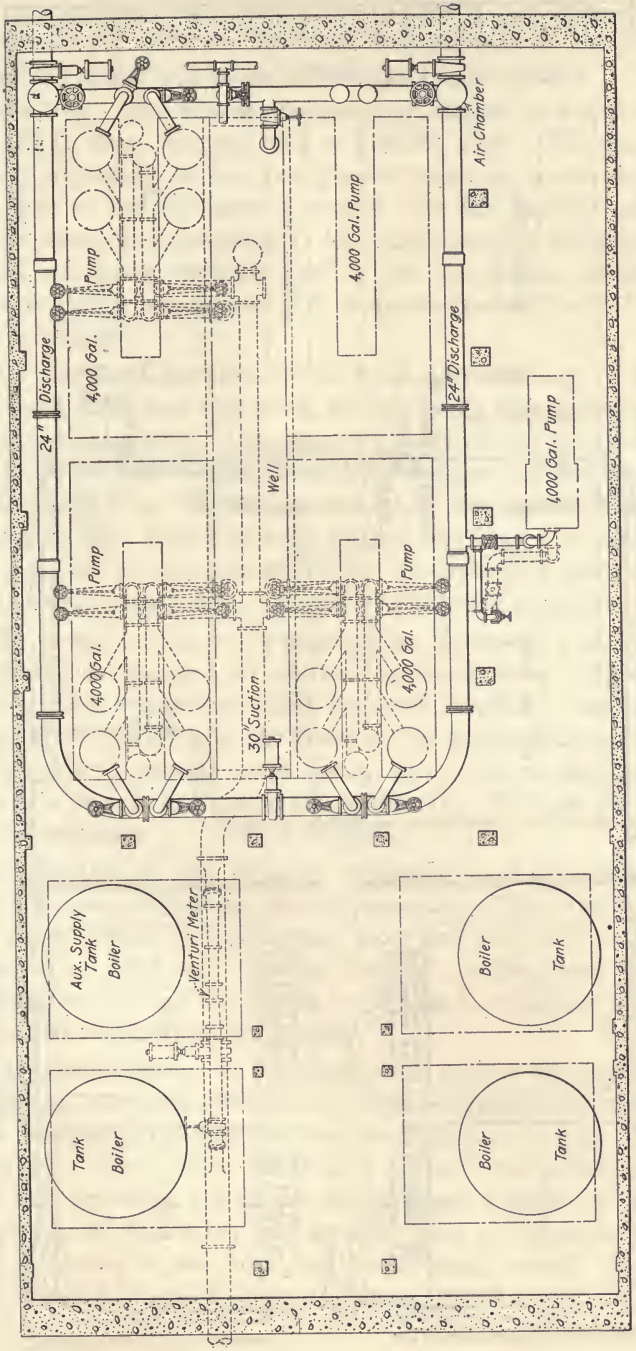


FIG. 18 PLAN OF STATION, SHOWING WATER PIPING

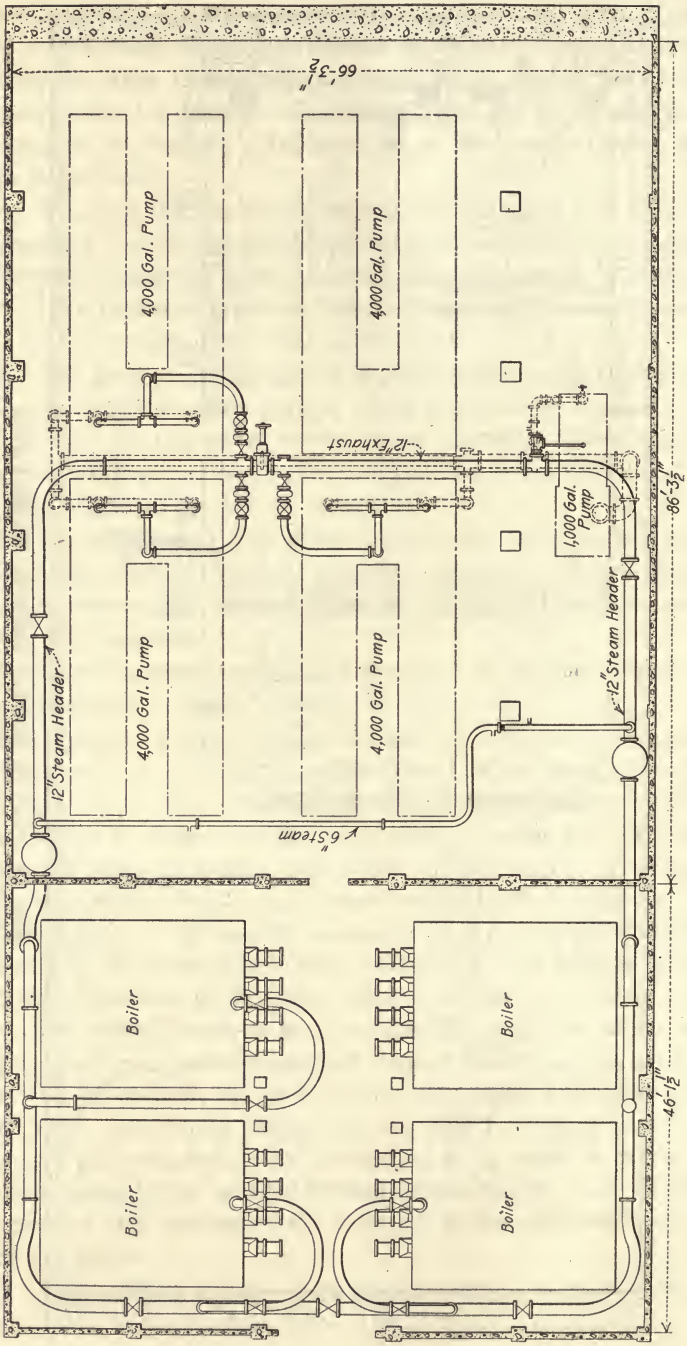


FIG. 19 PLAN OF STATION, SHOWING STEAM PIPING

pressure valve on the 18-in. riser serves to turn all the exhaust into the heaters at light loads, and provides a direct path to the atmosphere when the steam is in excess of the heater requirements.

90 *Building.* The site of the pumping station is a lot 69 ft. front by 137 ft. deep, running back to a 16 ft. alley. The property immediately adjoining on the two sides is occupied by warehouses carrying more or less inflammable stocks. On the opposite side of the alley the same conditions exist. All foundations for the pumping station and machinery were designed to rest on caissons carried to the gravel, so that it was necessary to underpin the walls of both the adjoining buildings.

91 All structural portions of the building, including columns, girders, beams, floor and roof slabs, and all walls except the front, are of reinforced concrete. The adjoining buildings on the sides are from 10 to 20 ft. higher than the station roof, so the roof girders, beams and slabs were designed to withstand the shock of falling walls in case of fire. The side walls are not less than 8 in. thick in any part, and have no openings whatever. The rear wall is the same, except that there is one door at the level of the boiler room floor.

92 For access to the men's quarters an automatic push-button electric elevator is installed, in addition to the stairway. Standard brass sliding poles are also provided for quick response to an alarm. The quarters are located over the front of the engine room, and include a dormitory, dressing room, bath, toilet and reading rooms. In addition, a private bedroom, bath, parlor and office are provided for the chief engineer.

93 In addition to the fireproof construction of the building, further fire fighting equipment for the protection of the station consists of a water curtain for the exposed front and rear, and two 8-in. standpipes, to be fitted with monitor nozzles. A dangerous fire in the immediate vicinity of the station could thus be effectively fought from the roof as well as from the ground.

94 *Signaling System.* In addition to the regular fire alarm circuit, a separate telephone circuit runs to the pumping station from fire alarm headquarters, fire department headquarters, and the chief's night quarters. This circuit connects to contacts for portable telephones in each fire-alarm box in the high-pressure district. In addition to the regular Morse key and sounder there are contacts for a telephone connection in each box, over the fire alarm circuit. Finally there is available the regular public telephone service.

CONSTRUCTION COSTS

Portable Equipment

2 automobile hose wagons at $5000	$10,000
8000 ft. 3 in. hose at $1	8,000
30 portable heads and regulators at $385	11,550
Total	$29,550

Pipe System

Material delivered Baltimore

Hydrants, 226 at $100	$22,600	
8 in. pipe, 7137 ft. at $2.35	16,700	
10 in. pipe, 28,229 ft. at $3.10	87,700	
16 in. pipe, 17,052 ft. at $5.25	89,600	
24 in. pipe, 1275 ft. at $10	12,750	
8 in. gate valves, 6 at $100	600	
10 in. gate valves, 193 at $130	25,000	
16 in. gate valves, 90 at $210	18,900	
18 in. gate valves, 2 at $300	600	
24 in. gate valves, 3 at $1,000	3,000	
Air and relief valves	200	
Low pressure gates, 2–30 in	500	
Suction pipe, 400 ft. cast iron, 30 in., at $4	1,600	
Steel air chambers, 2–30 in., at $500	1,000	
Venturi meter	500	
Cast steel specials	17,500	
		$298,750

Installation

Laying pipe, including placing valves, fittings, hydrants, etc.		
8 in. pipe, 7,137 ft. at $0.70	$4,996	
10 in. pipe, 28,229 ft. at 0.75	21,200	
16 in. pipe, 17,052 ft. at 1.15	19,600	
24 in. pipe, 1,275 ft. at 1.75	2,230	
Pump connections in station	6,000	
Laying 30 in c. i. suction	3,400	
Tapping 40 in. main	1,500	
Concrete valve boxes, 293 at $30	8,790	
Excavation, back filling and rubble paving		
41,318 ft. open trench, at $3.84	158,600	
12,375 ft. tunnel, at $4.08	50,400	
Improved paving, 6650 sq. yd., at $1.50	10,000	
Superintendence, use of tools, etc	50,000	
		$336,716
		$635,466

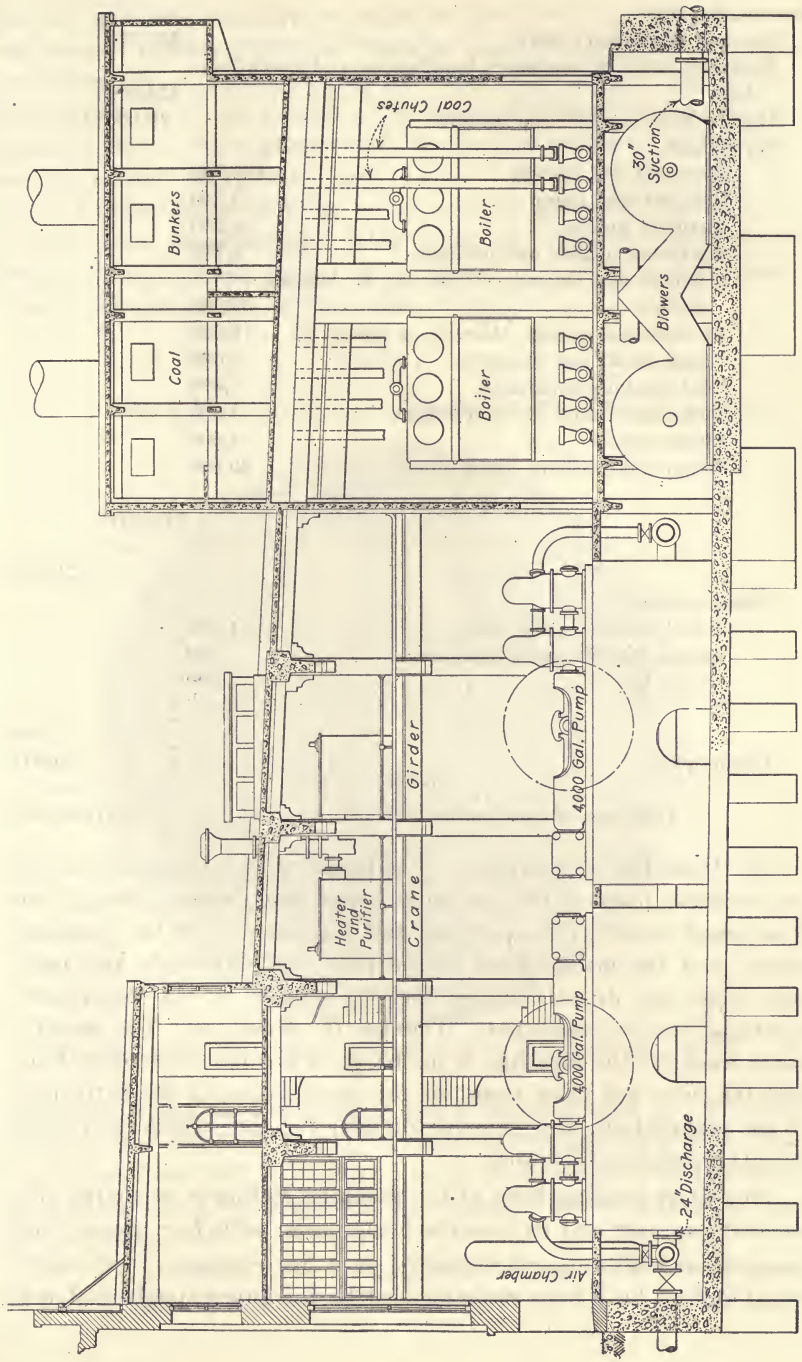


FIG. 20 SECTIONAL ELEVATION OF STATION

Pumping Station			
Site and preliminary work		$37,730	
Building, including machinery foundations, and men's quarters		124,800	
Harbor intake and screen chamber		10,000	
Equipment			
Four 4000 gal. pumps	$82,000		
One 1000 gal. pump	3,500		
Auxiliary pumps	4,250		
Feedwater heaters and purifiers	4,750		
4 boilers and settings, 27,200 sq. ft. heating surface	33,000		
16 underfeed stokers, blowers, air piping, etc.	18,000		
4 steel stacks and supports	8,000		
Coal handling apparatus	7,000		
Turbo-generators and switchboard	4,500		
Electric crane	4,000		
Steam and auxiliary water piping	30,000		
		$199,000	
			371,530
Miscellaneous			
Signal system, cables, etc	$1,500		
Furnishings for men's quarters	500		
Incidentals	5,000		
			7,000
Engineering			50,000
Total cost of construction			$1,093,546

95 *Operating Department.* The Board of Fire Commissioners is the executive head of the operating department, acting through the department chief. The operation and maintenance of the pumping station and the maintenance of the pipe lines, hydrants and portable heads are directly under the supervision of the department superintendent of machinery, Thomas H. Meushaw. The maintenance work for the pipe line is in charge of a general foreman, John Rudolph, who was chief inspector for the city during the early part of the installation, and general foreman for the contractor during the latter portion of the work.

96 The operating force at the pumping station is in charge of a resident engineer and five assistant engineers, with four stokers and two general assistants, all organized as a fire company. Two additional stokers have been recommended by the superintendent of ma-

chinery, and will probably be added in the near future, making a total station operating force of fourteen men. Of these an engineer and stoker are on active duty at all times, with a four-hour watch. Immediately upon the receipt of an alarm all hands report on the operating floor. As a pressure of 150 lb. is maintained during the standby period, orders have been issued that in the event of the pumps automatically speeding up without an alarm, the plant shall be shut down immediately. This course is taken to avoid water damage, should a break occur. Up to this time no occasion has arisen to require the execution of this order, however.

TESTS

97 *Opening Demonstration.* The system was formally placed in

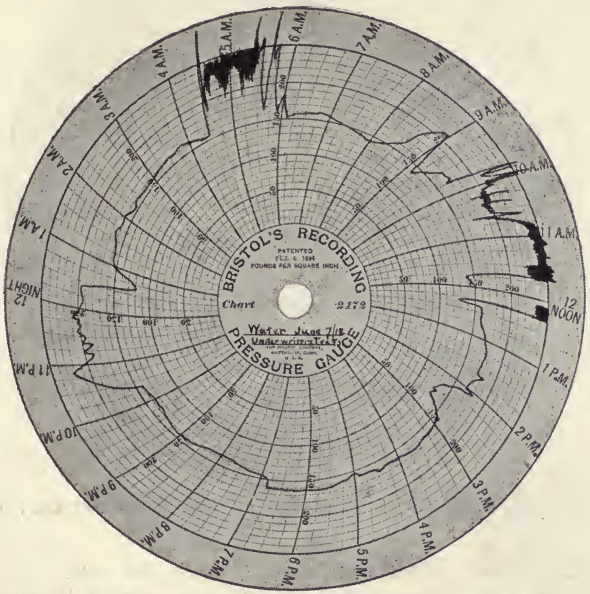


FIG. 21 CURVE OF WATER PRESSURE DURING UNDERWRITERS' TESTS

operation on May 20, 1912, with a public demonstration on the Court House Plaza. At first twenty-four 1¾-in. streams were used from single hose lines held by tripods. Later seven 2½-in. streams were thrown, from monitor nozzles on the wagons. About 13,000 gal. per min. were delivered.

98 *Underwriters' Tests.* A readiness test was made June 7, readings being taken by a stop-watch.

4.30.00 a.m. Fire alarm box pulled; one engineer and one stoker on duty at station.

4.31.00 a.m. Chief engineer with four additional engineers and three additional stokers had responded from sleeping quarters.

4.31.15 a.m. Two large pumps started up, pressure increased from 150 to 190 lb.

4.33.00 a.m. Pressure 280 lb. Hose company turned water on to two 3-in. hose lines siamesed into one 2½-in. monitor nozzle.

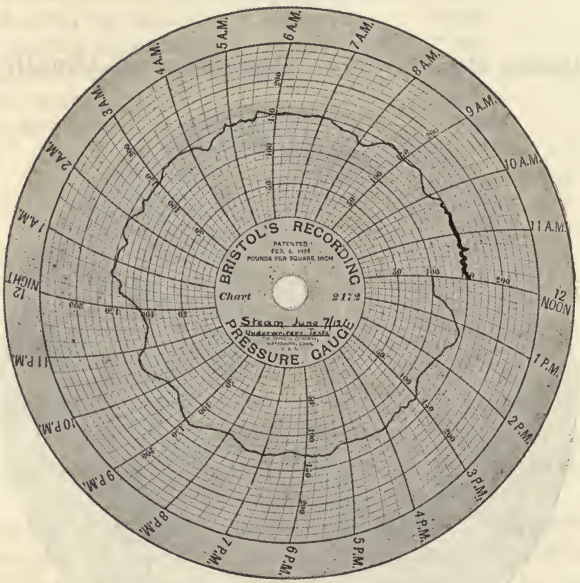


FIG. 22 CURVE OF STEAM PRESSURE DURING UNDERWRITERS' TESTS

4.34.00 a.m. Discharge was 1050 gal. per min.

4.37.40 a.m. Two additional 2½-in. nozzles in service, total discharge 4000 gal. per min.

99 A second company was then ordered into service at another point, with three 2½-in. nozzles, bringing the total discharge up to 7100 gal. per min. See water and steam charts from Bristol gage at station (Figs. 21 and 22).

100 A general performance test was made June 7, from 9.45 to 10.15 a.m. (see Bristol charts). Pumps were started and stopped, and hydrants were opened and closed as rapidly as possible to test

FIG. 23 VIEW OF STREAMS DURING UNDERWRITERS' TESTS

the ability of the pumps and governors to take care of abnormal operating conditions.

101 Individual pump tests were made the same date, from 10.15 to 10.45 a.m., with the following results:

Steam at throttle, lb	146
Water pressure, discharge, lb	199
Water pressure, injection, lb	41
Water pressure, net, lb	158
Revolutions per minute	55
Discharge, gal. per min	4592

102 A general capacity test was made 10.45 to 11.40 a.m., with the following results (three pumps in service):

Steam at throttle (average), lb	145
Water, discharge, lb	240
Water, injection	?
Water, net	?
Discharge, gal. per min	12,770

103 During the latter test six 2½-in. nozzles and six 2-in. nozzles were in use, discharging into the harbor (see Fig. 23).

RESULTS OF OPERATION

104 The department chief is enthusiastic over the efficiency of the entire system as a modern fire fighting equipment. He makes the statement that during the two months which it has been in operation, three bad incipient fires have been literally "drowned out" by the system. The probable losses from these fires, under former conditions, would more than have equalled the entire cost of the high-pressure service. By the time the department arrived on the scene at one fire, the smoke was so dense that from the middle of the street it was impossible to locate any window or door openings in the building. Three 2½-in. monitor streams were blindly turned on the building. The water reached the fire, but not through windows or doors. When the smoke had cleared away, three holes were discovered in the 18-in. brick walls, bored straight through the masonry by the high-pressure streams.

105 The engineers of the National Board of Fire Underwriters, after a thorough test and a special search for weak points, reported as follows:

> The distributing system has been installed for two years and shows no signs of deterioration. The slight leakage, absence of electrolytic action and total freedom from breaks or other troubles appear to justify the departure from the usual design of such systems. . . . The valve and hydrant distribution is

excellent, and the pipe sizes and gridironing are sufficient to enable a good concentration of flow without serious loss of pressure. . . . The separate hydrant head permits the use of regulator valves permanently attached, giving excellent control of the pressure on hose lines. The hydrant head under test showed sufficiently low friction loss. . . . The operation of the pumping plant is prompt and reliable.

106 As a result of the installation of the high-pressure system, the underwriters have announced a rebate of 5 cents in the insurance rate on all property in the district covered. While no exact summary has been made of the aggregate saving which will result from this reduction, it is roughly estimated that the amount will be approximately $40,000 per annum, which will be increased almost in direct proportion with the extension of the mains.

ORGANIZATION

Construction. The executive head of the project is the Board of Fire Commissioners. To act with the chief of the department the board appointed a consulting engineer, D. B. Banks, who in collaboration with the writer, designed the system. After the general plans had been drawn, but before the construction had been begun, the board employed two additional consulting engineers to pass upon the general features of the design. R. C. Carpenter and Frederick H. Wagner, chief engineer of Bartlett & Hayward Company, Baltimore, were chosen, and after a careful study these engineers approved the general plans and the details as far as completed. The architectural features of the station were designed by Henry Brauns, the veteran power plant architect of Baltimore. General supervision of the construction was exercised by Wm. McCallister, Jr., assistant to the consulting engineer. Before the construction work was completed, the department chief, George W. Horton, was retired, and the deputy chief, August Emerich, was promoted to the head of the department.

Contracts for the various elements of the system were awarded to the following builders and contractors:

Automobile hose wagons to the Mack Manufacturing Company.

The portable hydrant heads and regulators, and the main pump governors to the Ross Valve Manufacturing Company, of Troy, N. Y.

The hydrants and high pressure water pipe lines to the Pittsburgh Valve Foundry & Construction Company of Pittsburgh. Many of the working details of the system were designed by J. Roy Tanner, chief engineer, and Charles Fitzgerald, superintendent of construction for the contractor. A subcontract for the supervision of the trenching was awarded by the general pipe contractor to E. Saxton, of Washington, D. C., one of the most experienced contractors in the vicinity on subsurface structures in city streets.

Pumps to the Allis-Chalmers Company, of Milwaukee, although this concern was not the lowest bidder. A subcontract for the 1000-gal. direct acting pump and the boiler feed pumps was awarded to the Epping Carpenter Company, of Pittsburgh.

Boilers to the Edge Moor Iron Company of Wilmington, Del. A subcontract for the stokers, blowers and regulators was awarded to the Underfeed Stoker Company of America, Chicago.

The steam and auxiliary water piping in the pumping station to the Crook-Kries Company, of Baltimore.

The station building and harbot intake to the B. F. Bennet Building Company, of Baltimore.

The signal system was installed by the department force.

No. 1393 *c*

ALLOWABLE HEIGHTS AND AREAS FOR FACTORY BUILDINGS

By Ira H. Woolson, New York

Member of the Society

In the design of factory buildings, one of the vital features tending to control the spread of fire is a judicious limitation of height and area. It is self-evident that whatever restricts a fire reduces the life hazard. Owing to the supreme importance of these two subjects, a person contemplating the erection of a building of this class should give careful consideration to the history of fires in such buildings, and the experience gained in fighting them. The question is more acute in this class of buildings than in any other because of the fire hazard which exists in them, and the economic advantages due to reduced costs in construction and supervision, when several large areas are housed under a single roof. Just where to draw the line so as to produce reasonable safety without prejudice to building investments is the problem.

2 Factory buildings of excessive heights or areas have long been recognized by underwriting organizations as a grave danger to life and property, owing to the difficulty of controlling fires in them. They have for years urged limitations which have been freely ignored by ambitious architects and factory owners, because the suggested restrictions were considered unreasonably drastic. The evidence produced in this paper strongly supports the limitations which were advocated.

3 It is logical to assume that the men best fitted to determine safe limits of heights and areas are the men who have made a life work of combating fires under all conditions of weather and hazard. With this idea in mind, the writer communicated with all the fire marshals and fire chiefs in the United States representing cities of over 20,000 population. A set of eight questions and a letter of explanation were sent to each. Fire chiefs as a class are not good technical correspondents, therefore it was not surprising that only

Presented at the Spring Meeting, Baltimore 1913, of The American Society of Mechanical Engineers.

one-third of the men addressed responded to the appeal. However, replies were received from 117 representative cities well distributed as to size and geographical location. These have been summarized and form the basis of this paper. A few of the replies indicated a misunderstanding of the questions, and these were discarded. The questions were as follows:

1 What should be the greatest height allowed for manufacturing or warehouse buildings *without sprinkler equipment?*

Brick and joist construction.....Height in ft.—— or No. of Stories——
Fireproof construction.....Height in ft.—— or No. of Stories——

2 Take the same question as No. 1, but assume the buildings to be fully equipped *with automatic sprinklers.* What height would you approve?

Brick and joist construction.....Height in ft.—— or No. of Stories——
Fireproof construction.....Height in ft.—— or No. of Stories——

3 What should be the greatest floor area allowed in the same class of buildings *without sprinkler equipment?*

Brick and joist construction Area in sq. ft.—— or Width—— ft Length—— ft.
Fireproof construction.....Area in sq. ft.—— or Width—— ft. Length—— ft.

4 If the same buildings were fully equipped *with automatic sprinklers* what area would you approve?

Brick and joist construction Area in sq. ft.—— or Width—— ft. Length—— ft.
Fireproof construction .....Area in sq. ft.—— or Width—— ft. Length—— ft.

4 Each building was assumed as a good one of its class, with enclosed stairways and elevator shafts; and the chiefs were requested to base their answers upon experience in fighting fires in the class of buildings described, and to assume restrictions which would afford a reasonable chance of controlling a fire on any floor.

TABLE 1 GENERAL AVERAGE OF 99 TO 111 REPLIES RECEIVED FROM ALL CLASSES OF CITIES[1]

Type of Building	Stories in Height	Area between Fire Walls in Sq. Ft.
Non-fireproof, not sprinklered.....	3.1	6,300
Fireproof, not sprinklered.....	4.9	12,300
Non-fireproof, sprinklered.....	4.6	12,800
Fireproof, sprinklered.....	7.0	27,100

Average story height was 12 to 13 ft.

5 Naturally, and quite properly, the replies reflect the local conditions, such as the efficiency of the fire department, the water pressure, the combustibility of the goods being manufactured, the

[1]The variation in the number of replies (averaged) resulted from some incomplete answers.

number of sprinkler equipments in service, and the degree of congestion among the buildings. However, all conditions were represented, and the summary of so large a number of opinions should indicate fairly well the average condition throughout the country. (See Table 1.)

6 The answers regarding allowable heights were much more uniform than those relative to area. It is significant that 83 per cent of the replies would limit the height of a fireproof sprinklered factory building to less than ten stories. The opinions in reference to height of the other classes of buildings were exceedingly uniform, and consistently low.

7 Replies as to permissible areas in sprinklered buildings were widely divergent, but for the unsprinklered classes they were more uniform than would naturally be expected considering the great diversity of conditions under which they were prepared.

8 It is evident from the figures given, that the fire chiefs have no settled policy among themselves as to the credit that should be given to an automatic sprinkler equipment as a fire extinguishing device. A few enthusiasts would permit unlimited area in a sprinklered building, while on the other hand a considerable number would give very little or no increase, when sprinklers are installed. Two chiefs stated that their unfortunate experiences with sprinklers had caused them to lose faith in their reliability. As a whole, however, they are strongly in favor of sprinklers and are inclined to permit overgenerous areas in buildings so equipped.

9 In order that the replies may be intelligently interpreted they have been separated into three groups, Tables 2, 3, 4, according to size of the city represented, and each group has been analyzed to show the character of the answers given to each question.

10 In the cases referred to by an asterisk, where no limits to areas were given, they were not included in the averages, but were counted in the columns giving the number of answers above the average. In each group it will be noted that about the same number of men gave high answers to all questions, the proportion being one-quarter to one-half of the number in the group. The uniformity of height limits, and the lack of it in the area limits, is very apparent in all groups. It will be noted that the largest area values are given in Groups I and II, comprising the smaller cities. This is significant, and needs explanation.

11 Occasionally the fire chief of a small city has experience which would abundantly qualify him to estimate properly the merits of fire-

proof construction and sprinkler equipments; more often, however, his city has meager protection of this kind, and consequently he has little

TABLE 2 GROUP I SUMMARY OF ANSWERS FROM 52 CITIES WITH A POPULATION OF 20,000 TO 50,000

Type of Building	Stories in Height			Answers above Average	Area in Square Feet			Answers above Average
	Average	Max.	Min.		Average	Max.	Min.	
Non-fireproof, not sprinklered	2.8	6	1	13	6,000	20,000	1,150	15
Fireproof, not sprinklered	4.4	10	2	24	12,600	60,000	1,150	15
Non-fireproof, sprinklered	4.1	8	2	17	12,300	*60,000	3,000	17
Fireproof, sprinklered	6.3	12	3	18	27,300	*180,000	5,000	20

* Four votes received in favor of "no limit to area" in this class.

TABLE 3 GROUP II SUMMARY OF ANSWERS FROM 23 CITIES WITH A POPULATION OF 50,000 TO 100,000

Type of Building	Stories in Height			Answers above Average	Area in Square Feet			Answers above Average
	Average	Max.	Min.		Average	Max	Min.	
Non-fireproof, not sprinklered	3.2	6	1	8	8,300	40,000	2,500	5
Fireproof, not sprinklered	5.2	10	1	6	14,800	60,000	2,400	4
Non-fireproof, sprinklered	4.8	10	3	5	16,300	75,000	1,500	5
Fireproof, sprinklered	7.7	20	4	5	36,300	200,000	4,000	5

TABLE 4 GROUP III SUMMARY OF ANSWERS FROM 36 CITIES WITH A POPULATION OF 100,000 AND OVER

Type of Building	Stories in Height			Answers above Average	Area in Square Feet			Answers above Average
	Average	Max.	Min.		Average	Max.	Min.	
Non-fireproof, not sprinklered	3.5	7	1	17	5,400	10,000	900	15
Fireproof, not sprinklered	5.3	9	2	18	9,800	22,500	2,400	10
Non-fireproof, sprinklered	5.0	10	3	15	11,300	22,500	900	13
Fireproof, sprinklered	7.5	12	4	16	19,400	*80,000	2,500	9

* Two votes received in favor of "no limit to area" in this class.

opportunity to judge of their efficiency, and it is not strange that he should be a bit extravagant in the credit he would give them.

12 The most rigid restrictions on area are found in Group III embracing the large cities. As fireproof construction and sprinkler equipments are common in most of our large cities, it is reasonable to assume that the fire chiefs of such cities would have had more experience with such methods of protection, and be better able to decide what increase should be given in the size of a building when such protection is provided, than their less experienced fellow officers in smaller towns. It is thought quite proper to assume their figures are more nearly correct and should be given the most weight.

13 Significant evidence in support of this argument is found in the fact that four chiefs who give no limit to areas in non-fireproof

TABLE 5 ALLOWABLE HEIGHTS AND AREAS IN FACTORY BUILDINGS

Type of Building	Stories in Height	Area between Fire Walls in Sq. Ft.
Brick and joist construction, not sprinklered.	3	6,000
Fireproof construction, not sprinklered.....	5	10,000
Brick and joist construction, sprinklered....	5	13,000
Fireproof construction, sprinklered.....	8	20,000

and fireproof sprinklered buildings are located in cities having a population of less than 50,000 in which there are few fireproof factory buildings or sprinkler equipments. On the other hand only two chiefs, in cities over 100,000 population, suggest a "no limit area" in a fireproof sprinklered building, and none approves such areas for non-fireproof buildings.

TABLE 6 AREAS IN FACTORY BUILDINGS

AVERAGE OF THE REPLIES OF 50 FIRE CHIEFS SELECTED FROM 117, THE TOTAL NUMBER RECEIVED AS BEST QUALIFIED BY TRAINING AND EXPERIENCE TO PASS JUDGMENT ON THE QUESTIONS INVOLVED

TYPE OF BUILDING	STORIES IN HEIGHT	AREA BETWEEN FIRE WALLS IN SQ. FT.
Brick and joist construction, not sprinklered	3.2	5,200
Fireproof construction, not sprinklered	5.3	9,300
Brick and joist construction, sprinklered	4.8	10,500
Fireproof construction, sprinklered	7.5	21,600

14 With these thoughts in view, Table 1 has been changed somewhat to be more in accord with the weight of evidence. It is believed, therefore, that Table 5 represents more correctly the consensus of opinion among the fire chiefs of the country best qualified

to judge as to what should be the proper limits of height and area for factory buildings.

15 These values might be increased somewhat under the influence of especially favorable local conditions, as previously explained, but the writer submits that as they represent the average deliberate judgment of such a large body of men, so well qualified to estimate the hazard which the values involve, they should be given careful consideration, and should be increased only with the utmost caution.

In selecting the above replies, attention was specifically given not only to the personality of the fire chief, but also to the character and number of factory buildings in his city, and the probability of his having experience with both fireproof construction and sprinkler equipment.

The chiefs selected were distributed according to size of cities as follows: 32 from cities with a population of over 100,000; 14 from cities with a population of 50,000 to 100,000; 4 from cities with a population of 20,000 to 50,000.

It will be noted that the figures in this table of actual averages compare very closely with those given in Table 5 which was compiled by a somewhat arbitrary method in an effort to bring out the same facts.

EXTRACTS FROM FIRE CHIEFS' LETTERS

16 The following extracts from letters received from different fire chiefs in connection with this investigation may be of interest as indicating their attitude of mind in relation to the questions asked:

"In my opinion, from a fire-fighting standpoint, *no building* should be built over eight stories."

"In our city there is room to grow on the ground without building high in the air. It is almost impossible for a public fire department to fight a fire from the outside above 75 ft."

"The figures given mean that every 66 ft. by 66 ft. should have a brick wall through length of building with Underwriters' doors, same to be double. As for width, in no case over 66 ft. wide; with solid wall, same to reach above roof at least 6 ft. *Build on ground not in air.*"

"A building 8 or 10 stories high, out in the open where it can be attacked from all sides should be handled very readily by a modern equipped fire department."

"I think that a factory should never be more than four stories high. I almost feel that there is no such thing as fireproof construction from my own experience. I know that it is possible to store enough material in any building to burn it. I am very much in favor of dividing rooms in factories with fire-resisting walls, provided with automatic fire doors."

"While fireproof construction is the best, it is the contents placed therein that is the hazard to life and property. Buildings should not be constructed

to a greater height than can be reached by fire department ladders; 85 ft. to upper windows.''

''In my opinion no warehouse building ought to be over one story in height. In regard to manufacturing buildings, I will say that I do not approve of any of these buildings being over three stories in height. If they want room, let them build in length and not so high; that is just what makes such bad fires. These buildings have all kinds of combustible material in them and they are sure to jump to another building if they are four or five stories in height.''

''It is my opinion that all buildings for manufacturing and warehouses should be sprinklered, and not built higher than what the water supply will furnish and cover.''

''Do not think any fire department can successfully fight a large fire over six stories high, and ten stories allowed only when there are two sources of water supply with good pressure.''

''Area of sprinklered and unsprinklered buildings should be about the same, on account of increase in height allowed for fireproof buildings.''

''All buildings of character named should be sprinklered.''

''Joisted brick construction should not be allowed without sprinklers.''

''I think a good sprinkler system is one of the best fire preventions that has been invented in a great many years, and if kept up properly, it is pretty hard for fires to get away.''

''If I had my way I would not allow any manufacturing plant to do business until it were properly sprinklered. It does things when they should be done.''

''My experience with the 28 factories in this city has been that the sprinkler systems are out of order much of the time. Not looked after properly.''

''This department has had no unfortunate experience with the sprinkler system, but, I do not feel inclined to depend upon them.''

''The reason for not showing more favor to sprinklered risks, is because our experience with sprinkler systems in this city has shown them to be unsatisfactory, and not to be depended on.''

''Stairs should be of steel without any wood sides; if any wood in the construction then there should be sprinklers. Should be sprinklers in all elevators even if they are enclosed, for an elevator is a bad air shaft. Brick factories cut up with wooden partitions are generally hard fires to fight.''

''I do not approve of small rooms in factories, they make it very hard for a fireman to fight his way through smoke trying to find a fire when a building of this kind is partitioned off so much.''

''In considering the limiting of height and area of a building, the question of accessibility should play an important part.''

No. 1393 *d*

THE PROTECTION OF MAIN BELT DRIVES WITH FIRE RETARDANT PARTITIONS

By C. H. Smith,[1] Boston, Mass.

Non-Member

The importance of safeguarding stairways by placing them in towers well cut off from the remainder of the building and of protecting the openings made by elevators through the floors has long been recognized. Today more than formerly, these features are taken care of in the design of manufacturing buildings, including also well arranged towers for the main belts or ropes where this method of driving is employed. Fig. 1 shows how these features may be taken care of in a textile mill.

2 The following remarks apply more particularly to the older manufacturing buildings and to those of more recent construction where the best principles of design of stair and elevator towers and belt and ropeways have not been followed. Neglect to safeguard vertical openings through floors has resulted in serious loss of life among occupants of the building, who found themselves cut off from their accustomed exits by the rapid spread of fire up through such unprotected openings.

3 In mills insured with the Mutual companies stairs and elevators have generally been well arranged, and the fire protective devices such as automatic sprinkler systems, etc., have shown their value not only in reducing the loss of property by fire to a minimum, but also it has been demonstrated that approved construction, high standards of general order and neatness and efficient fire protection works as well to safeguard the lives of operatives employed.

4 At the present time there are approximately 1,500,000 people employed in the 2800 industrial works insured with the Mutual companies, located in 29 states of the Union and Canada. Since the inception of the system in 1835, there have been but 32 deaths caused directly by fires in these properties and 21 were in a fire in an un-

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sprinklered mill in 1876 before sprinklers were in general use. This would indicate that under present conditions, the loss of life would average less than 1 per year per 1,000,000.

5 Of the total of 32 lives lost, poorly constructed beltways which allowed the rapid spread of smoke and flame were to a large extent responsible for the deaths of 25 persons. The need of safeguarding the vertical openings through floors around the main driving belts

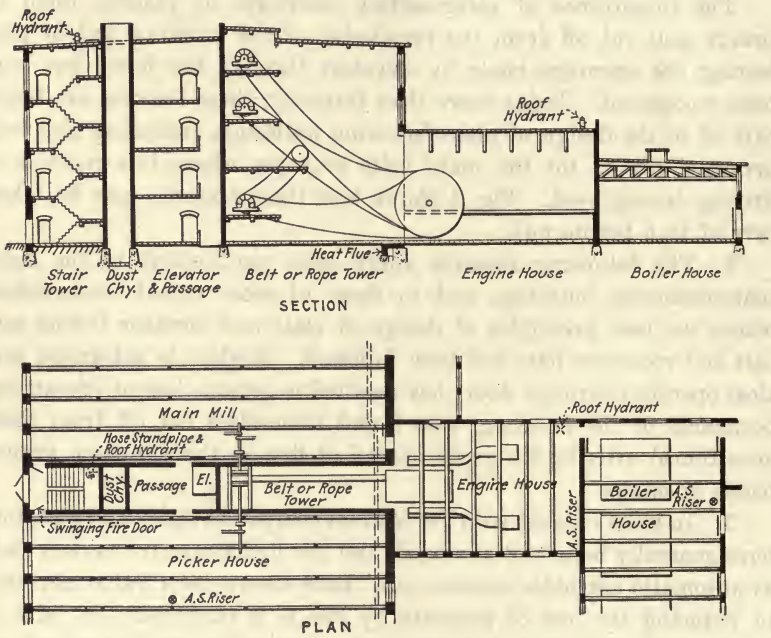

FIG. 1 BELT, STAIRWAY AND ELEVATOR TOWERS

had been less fully appreciated. Conditions at these drives were aggravated moreover, because it was the general custom to enclose the belts with boxes of wood, which in some cases were about head high and in others extended to the ceiling. The boxes tended to become oil soaked and to accumulate lint. A fire once starting at or near them would rapidly make headway, being carried by the natural draft up through the mill. Such a fire would also be more or less sheltered from the action of the sprinklers in the room.

6 The recurrence of several large property losses from this source led to consideration of this matter and measures were taken which

have to a great extent eliminated the open beltway hazard from Mutual risks. In the experience of these companies there have been about 20 fires occurring in the vicinity of main drives in which the open beltway was an important factor in the spread of the fire. These 20 fires resulted in a total loss of $2,721,635, an average of $136,082 per fire. Some of the larger of these losses occurred in the days before sprinkler protection was as complete as now, but the statistics showed that even with complete protection the open beltway was a serious hazard.

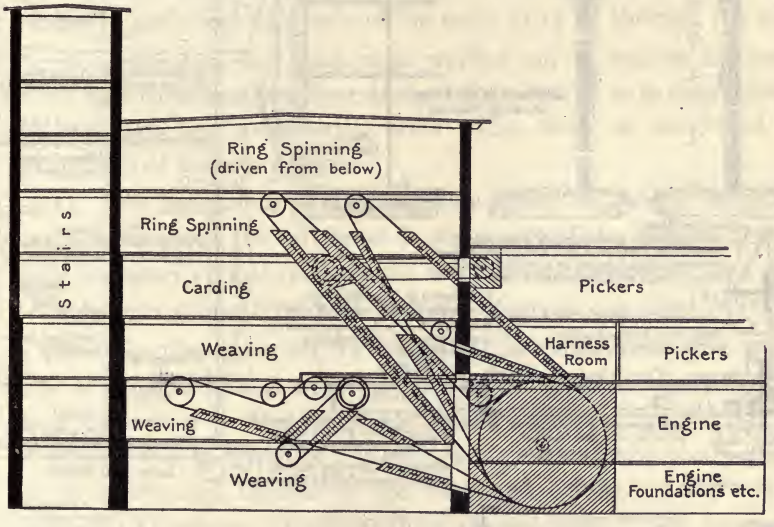

FIG. 2 SECTION SHOWING BELTS AND WOODEN BOXING BEFORE FIRE OF SEPTEMBER 15, 1907

7 The last bad fire from this source occurred September 15, 1907, at a cotton manufacturing establishment in Fall River. This is a stone mill, 339 ft. long, 74 ft. wide and five stories and basement in height with a 4-story wing, 94 ft. long and 65 ft. wide, projecting from the rear at the center of the mill. The engine room was located in the first story of this wing. The belts were boxed with wood and most of these were cut off head high in the several stories. Fig. 2 shows the general arrangement of the drive.

8 Sunday forenoon a bearing in the beltway just above the flywheel was being repaired. While the man doing the work stated that he had no knowledge of anything that could cause the fire, it is

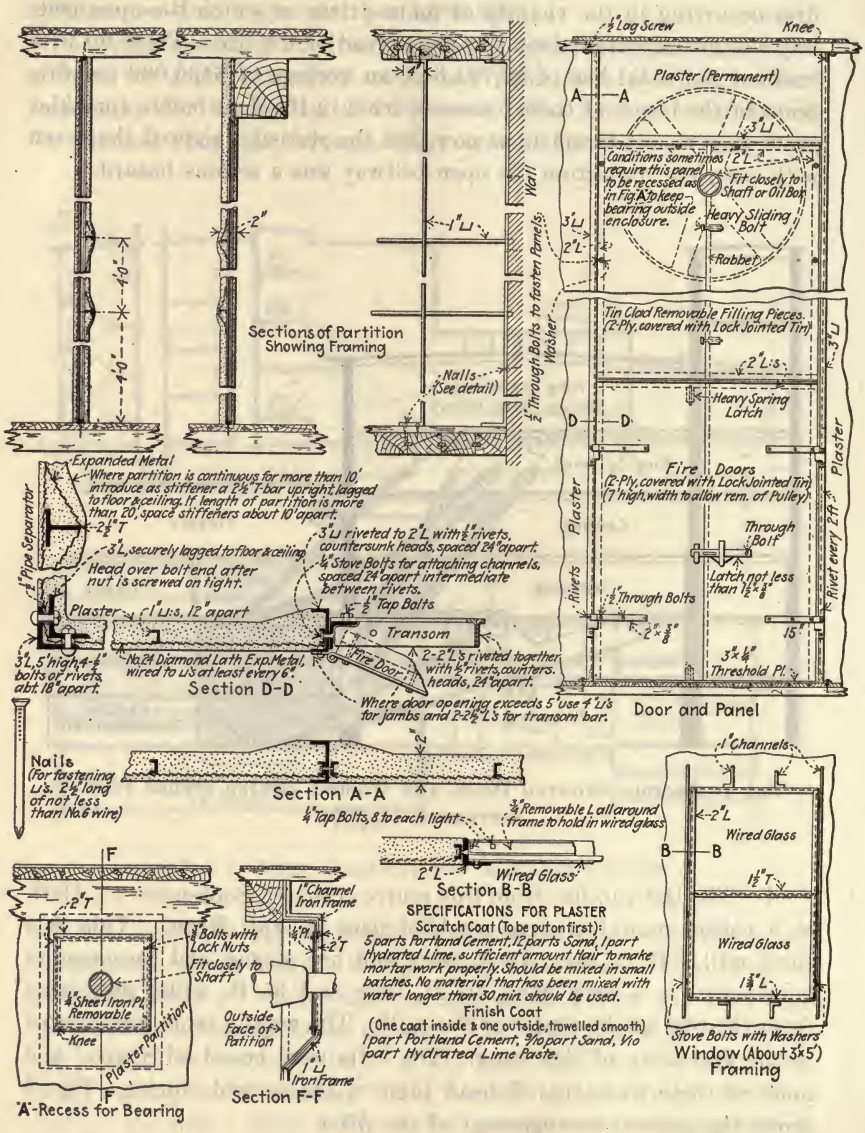
Sections of Partition Showing Framing
Wall
Nails (See detail)
½" Through Bolts to fasten Panels
Washer
Expanded Metal
Where partition is continuous for more than 10' introduce as stiffener a 2½" T-bar upright lagged to floor & ceiling. If length of partition is more than 20', space stiffeners about 10' apart.
1" Pipe Separator
3"L, securely lagged to floor & ceiling
Head over bolt end after nut is screwed on tight.
Plaster
1"L's, 12" apart
3"L, 5' high, 4-½" bolts or rivets, abt. 18" apart.
No. 24 Diamond Lath Exp. Metal, wired to L's at least every 6"
Section D-D
3"L riveted to 2"L with ⅜" rivets, countersunk heads, spaced 24" apart.
½" Stove Bolts for attaching channels, spaced 24" apart intermediate between rivets.
½" Tap Bolts
Transom
Fire Door
2-2"L's riveted together with ½" rivets, counters. heads, 24" apart.
Where door opening exceeds 5' use 4"L's for jambs and 2-2½"L's for transom bar.
Nails (For fastening L's. 2½" long of not less than No. 6 wire)
Section A-A
¼" Tap Bolts, 8 to each light
¾" Removable L all around frame to hold in wired glass
2"L
Wired Glass
Section B-B
1" Channel Iron Frame
Bolts with Lock Nuts
Fit closely to Shaft
4" Sheet Iron Pl. Removable
Knee
Plaster Partition
Outside Face of Partition
1"L Iron Frame
Section F-F
A-Recess for Bearing
SPECIFICATIONS FOR PLASTER
Scratch Coat (To be put on first):
5 parts Portland Cement, 12 parts Sand, 1 part Hydrated Lime, sufficient amount Hair to make mortar work properly. Should be mixed in small batches. No material that has been mixed with water longer than 30 min. should be used.
Finish Coat
(One coat inside & one outside, trowelled smooth)
1 part Portland Cement, 3/10 part Sand, 1/10 part Hydrated Lime Paste.
½" Log Screw
Knee
Plaster (Permanent)
Conditions sometimes require this panel to be recessed as in Fig. A to keep bearing outside enclosure.
Fit closely to Shaft or Oil Box
Heavy Sliding Bolt
Rabbet
Tin Clad Removable Filling Pieces. (2-Ply, covered with Lock Jointed Tin)
Heavy Spring Latch
Fire Doors (2-Ply, covered with Lock Jointed Tin) (7' high, width to allow rem. of Pulley)
Through Bolt
Latch not less than 1½"×⅜"
½" Through Bolts
3"×⅛" Threshold Pl.
Plaster
Rivets
Rivet every 24"
Door and Panel
1" Channels
Wired Glass
Stove Bolts with Washers
Window (About 3'×5') Framing

FIG. 3 DETAILS OF CONSTRUCTION FOR FIRE RETARDANT BELT ENCLOSURES

probable that its origin was connected with his work. After completing the job he left the locality. On returning 10 minutes later, he saw fire just below where he had been at work, and gave the alarm.

9 The fire passed up through the wooden belt boxing into all stories as far as the fourth floor where the drive terminated. The mill filled with heat and smoke so rapidly that in 5 minutes no one could enter the rooms. This was in spite of 650 sprinklers which opened, but in justice to the sprinkler equipment, it should be stated that the water pressure at this mill was weak. A section about 50 ft. wide was badly burned on each side of the main drive up through the mill.

10 After this fire plans were worked out to enclose the main drives with partitions of a fire retardant character, so as to approximate the standard belt tower with brick walls, such as are found in many mills of modern design.

11 The limitations of cost, available space, etc., which prevail in many places where the belt tower is not a part of the original design, make necessary special construction such as was adopted in this case, and has been successfully used in many others of the older mills.

12 The plan provided for inclosing the main drives with partitions of expanded metal and cement construction from 2 in. to 2½ in. thick depending on the story heights. A framework is constructed of expanded metal wired to 1 in. or 1¼ in. channel iron studs spaced 12 in. apart, and secured to the floor and ceiling. Longitudinal stiffeners of the same material as the studs are used. Where necessary, as in the case of a continuous partition of more than 10 ft., additional stiffness is secured by providing 2½ in. tee-bar uprights. On the frame so constructed portland cement mortar is applied by plastering to make a solid partition, all of the iron frame being embedded in the cement with the exception of the door jambs. These partitions, being comparatively light in weight, could be set up anywhere on the heavy mill floors without the necessity of strengthening them, although where possible it was arranged to have the partitions come over the beams. Although this form of construction for partitions has been largely used and with satisfaction, it would be possible of course to employ some of the special forms of studding now on the market which combine the studs and lathing in one sheet of metal. Details of the construction used are shown in Fig. 3.

13 While in general the enclosures occupy only the floor space necessary for the main belts, it was endeavored to have them as roomy as conditions of machinery installation would permit, in order to facilitate inspection and repairs to the main belts. Provision was made for taking down the lineshafting without disturbing the body of the partitions, usually by placing the fire doors which gave access to the enclosure under the lineshaft, and providing removable wood tin-clad panels constructed like fire doors above the latter. The main bearings were generally left outside the enclosures and to accomplish this the panels in front of the pulleys were sometimes recessed.

14 It was also the endeavor to arrange these enclosures so that they would be as well lighted as possible by including in them windows in the side wall of the building or providing wired glass windows in

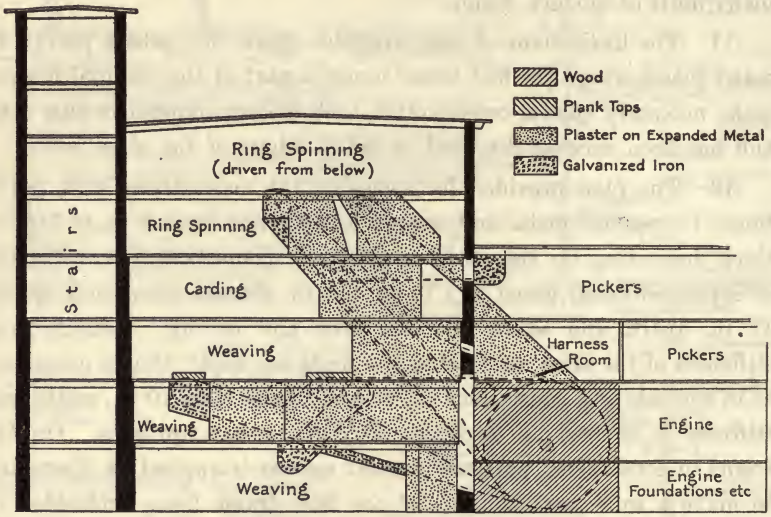

FIG. 4 SECTION SHOWING MAIN DRIVE AS NOW PROTECTED BY FIRE RETARDANT ENCLOSURES

metal frames to admit light to the beltway from the room. Fig. 4 shows diagrammatically the completed work at the Fall River mill, and Figs. 5, 6, 7 and 8 are photographs of belt enclosures in different stories. The adaptability of the construction is evidenced in the sloping sides and offsets which it was necessary to make in many cases on account of crowded conditions in the vicinity of the main belts.

15 While there is no claim that these partitions are as efficient

in withstanding the action of a severe fire as a brick wall would be, they are undoubtedly effective in preventing the dangerous draft up through an open beltway. In an actual fire in one of the mills where this construction was installed these enclosures were successful in confining the fire to narrow limits, and undoubtedly prevented a very serious loss.

16 *Stairways.* Where interior stairways are not properly enclosed in brick towers, it is possible to improve the conditions with enclosures of the same type of construction as employed in the beltway work, although it would be much better where the appropriation can be secured to build a standard tower of brick or concrete, especially if the mill is of any considerable height. Placing the stairs and elevators in towers projecting from the mill wall frequently results in a gain of valuable floor space.

17 The type of stair tower that has been developed in the factory buildings at Philadelphia is deserving of more general adoption as it combines with its functions of a stair tower that of a fire escape in the best sense. It consists essentially in a tower separated from the mill so that access to it can be had from the several floors of the mill only from an outside platform or from a vestibule which is open to air. Such a tower can never become filled with smoke from a fire in the mill. Many of the older mills in other sections of the country have stair towers that can be readily converted into towers of the Philadelphia type by closing the openings between the stair tower and the mill in the several stories and arranging for an outside platform in each story communicating from the mill to the tower.

18 *Elevator Enclosures.* We have also found the use of expanded metal and cement partitions practicable for enclosing elevator wells that were not properly protected in the original construction of the building, or where they have since been added. The necessary openings at such elevator shafts should be closed, preferably with wood tin-clad doors of the type which serve as safety gates as well. Where space does not permit of the installation of such doors, rolling steel shutters arranged to be automatically operative by the melting of a fusible link, as well as manually, can be used providing the hazards of occupancy are not excessive.

19 *Other Uses.* The average cost of partitions of the construction advocated is from 30 cents to 33 cents per sq. ft. These figures are for the work in place and include a contractor's profit. These partitions have been used with superior results and not greatly in-

FIG. 5 HARNESS ROOM, SECOND STORY, DIRECTLY OVER FLYWHEEL SHOWING PROTECTION OF BELTS LEAVING WHEEL

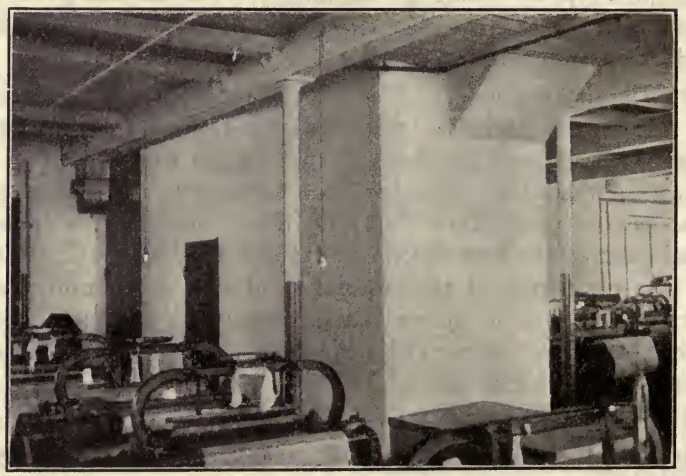

FIG. 6 WEAVE ROOM, SECOND STORY. NOTE FIRE DOOR WITH REMOVABLE PANELS ABOVE TO ALLOW ACCESS TO PULLEY ON LINESHAFT

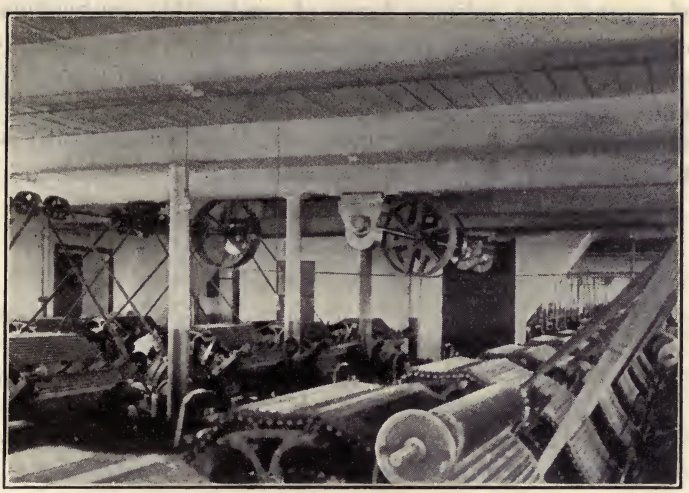

Fig. 7 Card Room, Third Story. Ends Sloped to Economize Space. Note Wire Glass Window and Fire Door with Removable Panels above

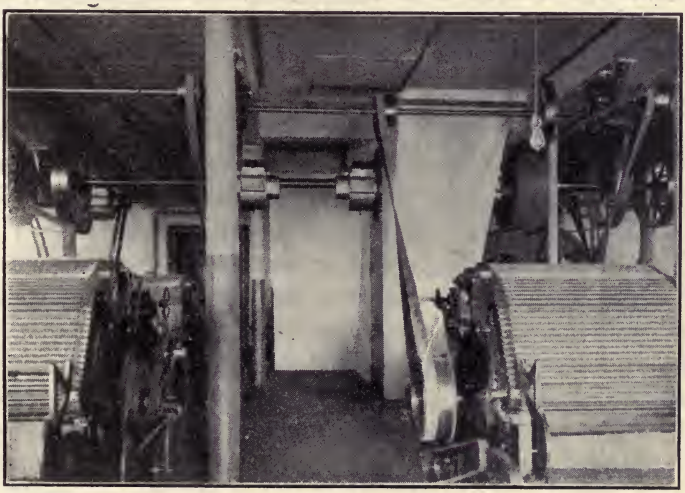

Fig. 8 Card Room, Third Story. End View of Belt Enclosure. Bearings All Outside

creased expense over ordinary forms of combustible construction for the purpose of separating special hazards from the remainder of a manufacturing room. For such purposes as the construction of bins to contain inflammable stock, the segregation of waste working machines, construction of lacquer rooms, etc., uses are constantly being found for this material in manufacturing works.

No. 1393 *e*

THE LIFE HAZARD IN CROWDED BUILDINGS DUE TO INADEQUATE EXITS

By H. F. J. Porter, New York

Member of the Society

Buildings in general are either non-fireproof or fireproof. The former can be compared to a pile of kindling wood out in the open, sometimes oil soaked and always ready to be set on fire; the latter to a stove full of fuel ready to be set on fire. In both cases the human occupants swarm around in the interstices in the pile of fuel, and as soon as the fire starts those caught in the fagots have to work their way down through the smoke and flames to the ground to safety.

2 Factory buildings in particular are sources of great danger to their large number of occupants, both on account of their non-fireproof construction and because of the obstructions to rapid egress, due to haphazard placing of machinery, furniture and partitions and the small number, size and character of the exit facilities.

3 Of late, the unrestricted use of fireproof construction in the buildings themselves has been advocated and the author has recommended the development of a form of exit drill of the occupants of each building to determine if, in the case of danger, they could escape readily from the building and if they could not, the alteration of the exits until they could. By "readily" is meant within three minutes, for from many conferences it was found that people do not want, nor would it be safe, to remain in a burning building longer than that time.

4 The capacity of a stairway, if time is not a factor and a stream of people pours into it only at the top and out of it from the bottom, is unlimited; but if time is to be considered the capacity is limited by its cross-sectional area. In a multi-storied building with crowds of people on each floor trying at different points in its length to get on to one stairway in a limited time, the conditions are very different. If more people try to get on to the stairs from each floor than the section between that floor and the floor below will

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hold, a jam will occur so that the flow downward will cease. The capacity of this section is very limited.

5 A crowd of people does not flow like a liquid composed of round smooth molecules. Their soft bodies are angular in shape more like pieces of rubber with wires in them and they therefore interlock. Clothes present rough surfaces causing friction and if the stairway is narrow an arch is apt to form across it which can become an obstruction in case of pressure from above such as actually to burst the stair rail or enclosing partition.

6 The capacity of a stairway of the average height of from 10 to 12 ft. between floors and not less than 22 in. wide would be one person to every other step or 10 and 12 per floor respectively, and if the width is doubled (not less than 44 in.) so that two people can come down abreast, twice those numbers or 20 and 24. If a stairway has winders in it, its capacity is reduced 50 per cent. One person can descend a single flight of such steps 10 to 12 ft. high in 10 seconds, striking a gait which he can maintain for seven or eight flights of steps. After that he goes slower, making the tenth flight in about 11 or 12 seconds. Every person added in single file adds 1 second to this time. A double file takes no longer if the stairs are double width. Thus it will take 10 seconds for 10 or 20 people, that is, the full capacity of a flight of steps, to come down one story. The capacity of a stairway may be thus increased by widening it in multiples of 22 in. A crowd of people cannot be depended upon to come down more than ten stories. One or more of them will give out, and demand the attention of others. Those who do get down will be severely taxed. The total time required to empty a building is determined by the time required to empty either the floor farthest from the ground or the floor occupied by the greatest number of people.

FORMULA FOR EMPTYING A FLOOR BY ONE STAIRWAY

Number of couples (number of people divided by 2)	c
Time of formation in line after signal, seconds	10
Time one couple takes to march to top of stairs, seconds	10
Time each couple takes to pass through door at top of stairs, seconds	1
Number of stair flights (one less than number of floors)	f
Time of one couple to descend one flight of stairs, seconds	10
Time of one couple to go from foot of stairs to street, seconds	10

$$\text{Total time} = T = 30 + c1 + f10$$

Example Time of emptying 100 people from tenth floor

$T = 30 + 50 + 90 = 170$ seconds $= 2$ minutes, 50 seconds

Example Time of emptying a ten-story building with 20 people on each floor is the same as emptying 20 people from tenth floor.

$$T = 30 + 10 + 90 = 130 = 2 \text{ minutes, } 10 \text{ seconds}$$

7 Tests of the capacity of fire escapes in a limited time gave the following results: A straight ladder, 2 per floor; ladder set at 50 to 60 deg. with the horizontal requiring people to go down backwards 3 to 4 per floor; stairs 30 in. wide, 10 to 12 per floor; and the modern outside stairway with a mezzanine platform 40 in. wide, 20 to 24 per floor, the same as an inside stairway. Fire escapes are usually so exposed to flames from windows opening upon them that they are more often fire traps than fire escapes. They should be prohibited by law and safer methods of escape provided.

8 In order to insure the safety of the occupants of a building in case of emergency one of two things has to be done: (*a*) there should be two stairways so that if one is cut off by flames or smoke the other can be used and the number of occupants reduced on each floor to meet the limited capacity of the part of the stairway between floors, or (*b*) the number of stairways increased so as to have two separate and independent stairways from each floor to the ground with its own exit from the building. People can then pour into the top of whichever one is not cut off by the fire and continue down and out at the bottom without colliding with those from any other floor. Fire drills installed under either of these conditions worked more or less satisfactorily, and the author tried unsuccessfully for years to have ordinances passed in New York City and legislation enacted at Albany, making them mandatory, but the expense of changes in the buildings and the idea of having employees walk out of a factory while manufacturing operations were under way, upon the sounding of an unexpected signal, did not appeal to factory proprietors as practical. It required holocausts in New Jersey, Pennsylvania and New York finally to bring about the legislation in those states.

9 As time passed, however, the author developed what might be termed an exit test in factories which presented the opportunity and found to his astonishment that almost without exception, exit facilities adequate for handling the regular number of occupants under emergency conditions, were lacking.

10 This situation has probably developed with the rapid growth of industry where a factory building had been built to accommodate a certain number of people, and then, as the business grew, more people were accommodated without realizing that each additional person became an increment of danger to all. Or, if the danger was

at all appreciated, some means of escape from windows was supplied which might be anything from a rope to a ladder. After this condition had become general it crystallized into custom, and new buildings with exit facilities inadequate for their occupancy were designed, erected and accepted as safe. Ropes were followed by ladders, and these in turn by fire escapes which became in time an established necessity.

11 Engineers, when called upon to supply a mechanism, are expected to have it subjected to a working test, which it must pass before they get paid for it; but architects and builders have never been called upon to demonstrate by actual test that the facilities which they have supplied in their buildings for the purpose of emptying them under emergency conditions will actually work, and this notwithstanding repeated instances of panic congestion on stairs, of people being burned to death on fire escapes, of elevators sticking from the warping of their runways from heat, etc.

12 When subjected to test these exit facilities in many buildings have been found to be entirely wanting in adequacy, and when this fact was brought to the attention of those who were responsible, it has been surprising to find how readily they accepted the criticism. On the other hand, those who possess these unemptiable buildings are sceptical of such statements and unwilling to be persuaded that the buildings are not safe. They point to all the other buildings erected by reputable architects and builders and naturally are incredulous.

13 In order to empty these buildings, additional stairways had to be built and fire drills developed to take the people out. Such changes in the building are expensive, for two stairways have to be installed from each floor to the ground, so that if one is cut off by a fire, the other can be used. In many-storied buildings the number of stairways required becomes impractical. In addition fire drills are expensive to operate, for they involve not only loss of time of operatives and a break in the continuity of the process of manufacture, but the actual going down stairs and return of people, some of whom may be lame, others affected by a weak heart or lungs, others anaemic or organically weak, reduce the efficiency of the working force for a very appreciable time. If the drill takes place at the end of the day this criticism might be modified slightly.

14 Such is the situation in the usual type of factory building to be found in the average town where ground is cheap, buildings large and stairways broad. Turning now to the loft building used for

factory purposes, the conditions as regards emptiability are found to be very much worse and have to be corrected in a different manner.

15 Let us consider for the moment a one-story or ground-floor factory building with a doorway at each side, one of which is cut off by a fire. The people can march out horizontally through the other doorway and nothing will inpede this horizontal exit except the size of the doorway. If this is 22 in. wide, a single file of people can pass out in an orderly manner at the rate of one person every second. If

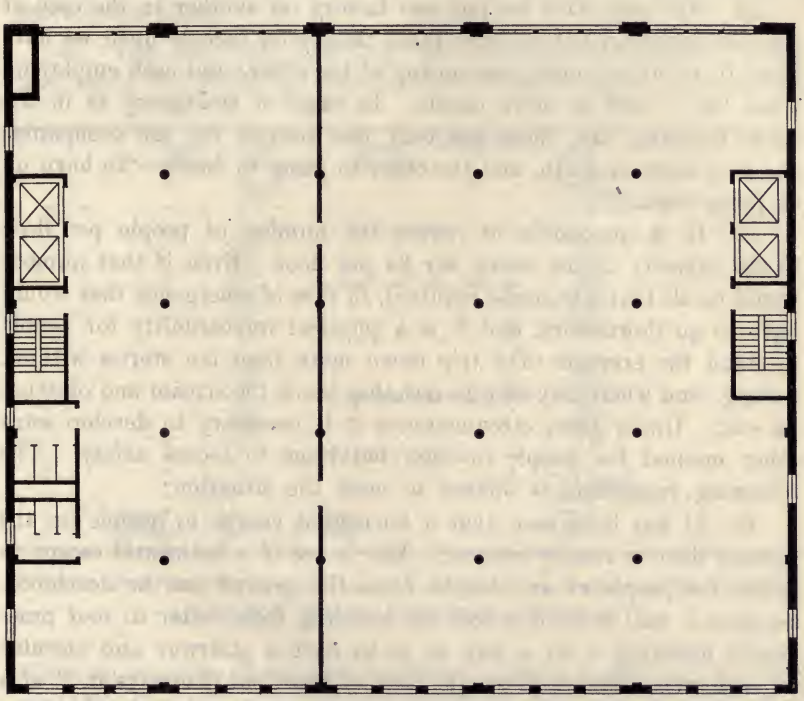

FIG. 1 FLOOR PLAN OF TYPICAL LOFT BUILDING SHOWING FIRE WALL WITH DOORWAYS

it is 44 in. wide, a line of people two abreast can pass out in the same time. One hundred people can make their exit through one 44-in. door, therefore, in 50 seconds, or say one minute.

16 Now put another factory on top of this one with one hundred people in it. The doorway at each side will have to open on stairways which lead down to the doorways constituting the exits from the factory below. Suppose a fire occurs on the floor below, cutting off one of these exits, the 100 people on the lower floor immediately

proceed to make their horizontal exit, while those on the upper floor proceed to make a vertical downward exit to reach the doorway out of which those below are moving. The result is of course a collision, the stream of people from upstairs coming down upon the stream of people on the ground floor on their way out. This collision prevents both the upstairs stream from coming down and the down-stairs stream from going out. There is a complete lock, and the building does not empty.

17 Not only have we put one factory on another in the case of our loft building, but we have piled factory on factory until we have from 10 to 30 and more, one on top of the other; and each employing from 100 to 300 or more people. In cases of emergency as in the Asch Building fire, there are only two courses for the occupants: one is to burn to death, and the other to jump to death—"to burn up or jump down."

18 It is impossible to reduce the number of people per floor to the capacity of the stairs, say 24 per floor. Even if that number would be all that a business required, in case of emergency they would have to go downstairs, and it is a physical impossibility for people to stand the exertion of a trip down more than ten stories without resting; and when they stop to rest they block the stream and obstruct its exit. Under these circumstances it is necessary to develop some other method for people in high buildings to secure safety. The following suggestion is offered to meet the situation:

19 It has been seen that a horizontal escape by people on the ground floor is readily secured. Let us see if a horizontal escape to safety for people at any height from the ground can be developed. Suppose a wall is built across the building from cellar to roof practically bisecting it in a way so as to have a stairway and elevator on each side. This wall should have at least two doorways in it at a considerable distance from each other and closed by self-closing fireproof doors (Fig. 1).

20 It is improbable that a fire will occur on both sides of this wall simultaneously. It could occur only by incendiary origin, and that would hardly be possible in working hours. Should one occur on either side, the people on that side would go through the doorways in the fire wall, close the doors after them and be perfectly safe. That half of the building in which the fire might be should be emptied in less than a minute if there were no more than 100 people on each floor to pass through one doorway 44 in. wide. If the principle of the horizontal escape presented by the fire wall is included

in the design of new buildings a most satisfactory method of securing safety at comparatively small expense will be offered.

21 In every way possible the horizontal escape should be developed in old buildings and the vertical escape subordinated. Factory buildings adjoining one another may have doorways through their sides connecting them on various floors closed by fireproof self-closing doors, or may be connected by outside balconies built around the party walls; or, if of different heights, doors in the sides of one may lead out on the roofs of the others.

22 The fire wall bisecting the building as described makes practically two buildings, each provided with elevators and stairways.

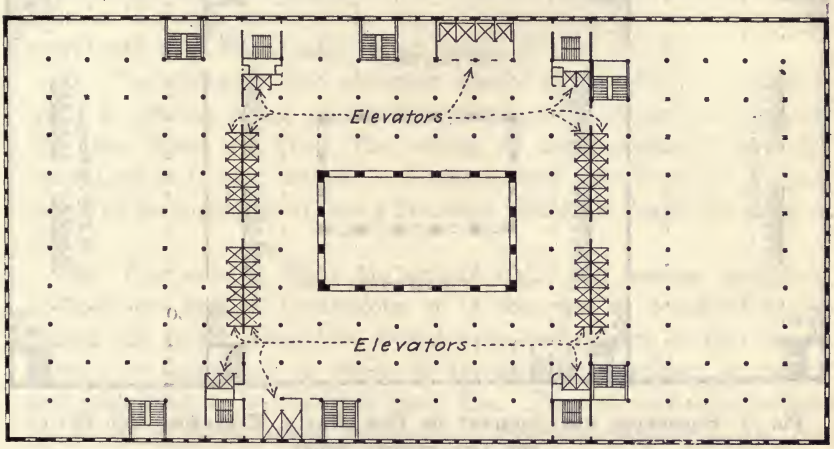

FIG. 2 DEPARTMENT STORE FLOOR PLAN SHOWING PRESENT ARRANGEMENT OF FIRE WALLS, ELEVATORS AND STAIRS

A fire on one side of the wall would be confined to half the building, and therefore the property loss would be reduced one-half. Only one-half the people would be endangered and have to move, and the distance they would have to go would be only one-half what it would be if they were on the ground floor of a building without a fire wall. They could remain on the same floor till the fire was extinguished, or could go down to the ground by the elevators operating under normal conditions.

23 The fire wall eliminates the necessity for a fire drill with its accompanying objections. Of course all buildings occupied by many people should have a fire alarm signal system in them to advise the people promptly of their danger. In buildings where there is a fire

wall the signals should be arranged so that in case a fire should occur on one side of the fire wall on any floor, a bell on each floor on the same side of the fire wall would ring, indicating on which floor the fire is. Then all the people on that floor and above it should pass through the fire wall and close the doors. Those below need not disturb themselves until the fire threatens them, and then they too can pass through the fire wall.

24 There are certain other safety devices which should be supplied in factories to protect the lives of the operatives from fire.

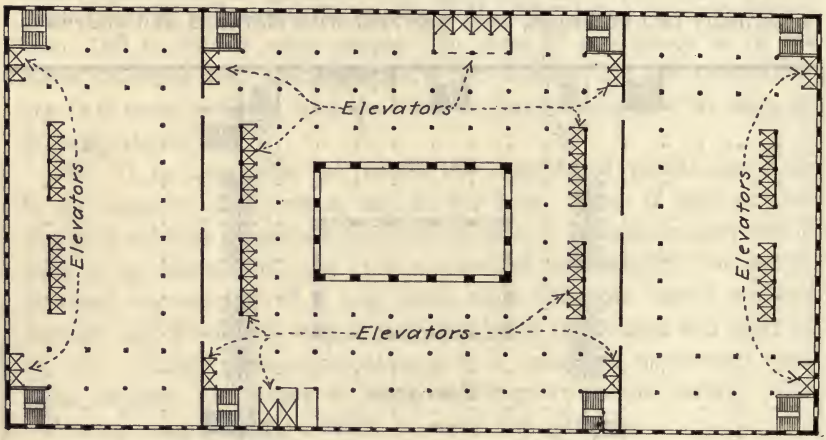

FIG. 3 SUGGESTED ARRANGEMENT OF FIRE WALLS, ELEVATORS AND STAIRS FOR DEPARTMENT STORE

One of these is metal-framed windows with wire glass. These are made so as to close automatically in case of fire, thus preventing the latter from spreading upwards from floor to floor outside of the building.

25 Another safety device is automatic sprinklers which serve to extinguish fires in their incipiency. All doors should be made to swing outward, and where they open on a hall or stair landing they should be vestibuled, so as not to obstruct the passage way. Sliding doors should be avoided if possible, as they are apt to stick or jam by pressure of people upon them.

26 Each floor of our typical loft buildings is say 100 ft. by 100 ft. by 10 ft. and therefore contains 100,000 cu. ft. of air. The laws of New York and many other states require 250 cu. ft. of air per person as a limitation of occupancy. This limits the number of

people per floor in a building of this size to 400 and if the stairways were 44 in. wide (and there are none now over 36 in.) at most only 40 per floor could possibly go down them even if the other 360 would let them.

27 With the fire wall only 200 of the 400 people on each floor would have to move, and if there were two doorways in the fire wall at some distance from each other, they could reach safety through them in one minute, or if one were cut off by the fire, all could pass through the other easily in two minutes. More doorways can be introduced, and thus the time of exit could be lowered still further.

28 . An effort is being made to increase the amount of air space required per person from 250 to 500 cu. ft., which would reduce the number of people per floor to 200, of whom only 100 would have to move, and they could easily reach safety in one minute.

29 The stairways and elevators should be enclosed in fireproof walls to prevent a fire on one floor continuing upward and setting the other floors on fire. The ceiling of the basement where the machinery is located should be fireproof, and should not be pierced inside of the building, so that a fire there would not reach the elevator shafts.

30 Fire escapes which are simply stairs and possess dangerous features not only of limitations as to size, but of accessibility for flames and smoke, should be looked upon as evidence of the incompetence or ignorance, or worse, of the architect, builder, or owner, and prohibited by law under a heavy fine. They are not only dangerous to life by giving a false confidence in their adequacy for escape, but they destroy the appearance of the building. Our cities should be built without such architectural blemishes.

31 Fire escapes of the chute type are tubes with a smooth helix instead of steps. If the only opening is at the top they have considerable capacity. They soon rust, however, and at best are not to be considered seriously in comparison with other means of safe exit. People cannot enter them at different floors while a stream of people is passing down from above.

32 The smoke-proof tower, claimed to have originated in Philadelphia, is the latest improvement in the line of fire escapes. It is simply an enclosed stairway on the outside of a building, but cannot be reached except by going out of doors. Its special claim is that smoke and flames cannot get into it. It has, however, no more capacity than any other stairway, and as its approach is always open to the weather and its interior is always more or less

dark, it is never used in ordinary service and becomes neglected. These monuments to architectural incompetency can be seen here and there filled with the dust and accumulated rubbish of every unused open space. When a time arrives for using them everybody has forgotten their existence. During the last year or two, notwithstanding the protests of many, a great many new buildings have been constructed, especially in New York City, with these monstrosities on them, and have been accepted by the building department in all seriousness.

33 The fire wall should be introduced into all buildings where the public congregates in large numbers. Large department stores, which on certain days are said to accommodate several thousand people per floor, are very dangerous places at present. A fire, or a panic without a fire, might cause a fearful tragedy. It is criminal for their owners to object to fire walls and offer as an excuse that they would obstruct the vista. Certain cities require fire walls in such buildings now as a property protection, and the vista is dispensed with without comment. The department stores of Philadelphia are so divided; John Wanamaker's new store there is divided by two such walls as shown in Fig. 2. The exit facilities in it, however, are badly arranged, for the architect apparently did not think of the life hazard of its occupants, and designed the fire walls to protect property only. Fig. 3 shows how the building might be redesigned so as to be safer. It should be noted that the elevators are removed from the fire wall so that people trying to go down in them would not block the doorways of the fire wall and prevent others coming through them. The stairways are situated as far from the fire wall as possible and should be enclosed by fireproof partitions.

34 Churches, assembly halls and similar ground-floor buildings should have their floor fireproof and unpierced so that any fire occurring in the basement would not endanger the occupants of the main building.

35 Moving picture buildings, theaters, etc., should be redesigned (Fig. 4). People come out of them by the way they go in, and in case of emergency all crowd into the narrow aisles. These aisles should be turned across the room and lead directly to courts opening on the street in a way such that streams of people will not collide. The various balconies and galleries should have foyers behind fire walls with separate stairs and street exits so that patrons will not have to mingle with those making their exit from the lower floors.

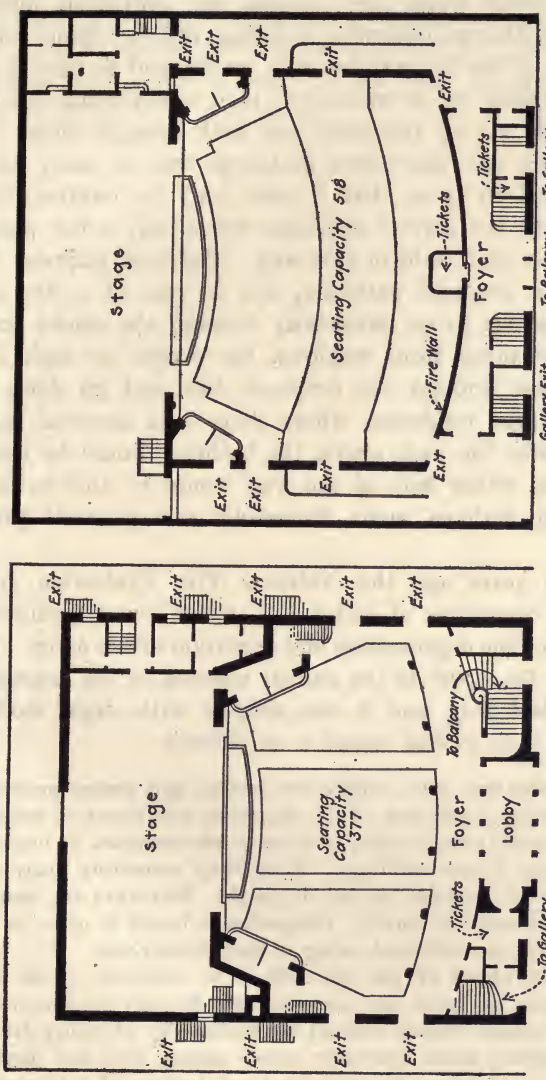

FIG. 4 TYPICAL AND PROPOSED THEATER PLAN SHOWING USE OF FIRE WALLS, SIDE ENTRANCES AND EXITS

36 Every school building should be divided by a fire wall providing a horizontal exit on each floor, so that the children will not have to be drilled to go downstairs in case of fire.

37 Hospitals where the inmates are bedridden, blind, lame, invalid, imbecile, or otherwise helpless, can be made safe by the introduction of the fire wall between wards, and in case of fire those who are bedridden can be wheeled on their beds through the doorways, and those who are up and about can walk through them.

38 Hotels and apartment buildings can so easily have a fire wall developed in them that it need only be referred to here in passing. Even the private residence where only a few pople occupy a floor can be made safe in this way. The back stairway should be enclosed in a fireproof partition, and in case of a fire instead of everybody having to go downstairs through the smoke and flames, or having to jump from windows, the people on each floor have simply to pass through the fireproof door and go down stairs in safety. In large residences where there is a servants' quarters in connection with the back stairs, the building would be bisected and the people on either side of the wall would be able to carry their clothing and perhaps much household and personal property to safety.

39 Two years ago the National Fire Protection Association appointed a committee of which the author was a member to draft suggestions for the organization and execution of fire drills. This committee made its report to the annual meeting of the association held in Chicago last May, and it was adopted with slight modifications. A prefatory note to this report is as follows:

> Many so-called fire drills, outside fire escapes, and similar practices and devices are generally insufficient, often dangerous, and therefore misleading substitutes for rational exit facilities, and are a manifestation of improper design and construction of our buildings. A stairway connecting many stories will accommodate only a limited number of people. Stairways are, therefore, dangerous means of exit for crowds. Congestion is bound to occur in them when used under stress of excitement owing to their limitations.
>
> The primary object of the exit drill is to determine if the building is properly designed so that in the emergency of a fire its occupants would be able to effect their escape readily without the probability of injury from stairway or other congestion which inevitably causes panic. This test should be occasionally repeated to insure the continuous maintenance of safe conditions.

40 The author advocates legislation, requiring three things: (*a*) Architects and builders should be prohibited from designing buildings which cannot be emptied within 3 minutes after a given

signal. (*b*) The municipal authorities should be required to institute an exit test in each building to determine, before it is accepted, if it can be emptied of its occupants in 3 minutes. If it cannot pass this test it will not be accepted and must be altered until it can pass the test. (*c*) Afterwards the proper authorities will be required to repeat the exit test from time to time, to see that the safe conditions originally established are maintained.

No. 1393*f*

DISCUSSION ON FIRE PROTECTION

W. H. KENERSON referred to the statement of Mr. Porter that people do not want, nor is it safe for them to remain in a burning building more than three minutes. He said that he was present at the start of a very quick fire where the people in the building were reluctant to stay at all. They got out of the building by jumping almost as quickly as they could run to the side walls. Even with adequate exits it was plainly evident that danger could not be eliminated. In buildings where operatives were working under crowded conditions some people would be burned or crushed before they could get out in case of a severe fire, even if the walls were open all the way around, owing to the furniture, machines, etc., being closely grouped. The panic following an incipient fire was often worse than the fire itself and did more damage. In some of our large cities, the streets in the neighborhood of office buildings and factories are so narrow and so hemmed in by the crowds rushing toward the building in case of a fire that there would not be room enough in the street to accommodate the operatives who were leaving. This was certainly true of some loft buildings in New York City.

Regarding the use of the fire wall, while it is of great value in certain cases and places, it is not altogether dependable. Was it not conceivable that fires could not start on both sides of a partition at once? There were many conditions under which a fire would rapidly spread to both sides of a partition even if it did not start on both sides, as for example in a flash fire.

The author very properly said that sliding doors inside of partitions were dangerous; but what of swing doors? These should always open outward, but what was "outward" in a partition that must be used in both directions? Where communication was established between adjacent buildings even, it was inevitable that one of the doors would open the wrong way.

There was only one paragraph in the paper about preventive methods through the use of automatic apparatus in conjunction with other things for preventing fire at the start. It was unfortunate that this was not included in a paper of the general scope of the one under discussion.

Presented at the Spring Meeting, Baltimore 1913, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

He believed that it would be impossible to enforce the legislation recommended in the last paragraph of the paper to prevent architects from designing buildings which could be emptied in 3 minutes. Before construction such a matter was a question of opinion rather than fact. Moreover, operatives could not be prevented from crowding or becoming panic stricken in the event of a fire and it was doubtful under such circumstances whether many a properly designed building could be emptied in 3 minutes even if it were possible to enforce the first recommendation.

HENRY HESS. I am neither an insurance nor a sprinkler man. I have had some experience putting up factory buildings and running the establishment afterwards. One of my most interesting experiences of that character was in building a large tool works in Germany. The first thing I ran afoul of was the law. I bring this up particularly because Professor Kenerson has indicated his belief that adequate legislation was absolutely impossible, if not as to securing it in the first place, then as to its later enforcement. Now that is a question of the will of the community. If the community really desires enforcement of law it will see to such enforcement. Fire wall partitions such as Mr. Porter advocates are very useful. In the case of the German factory partitions were installed in certain cases and were not insisted upon in others; but there was an insistence upon one other thing of possibly far greater importance. The authorities determined a central point for each floor area, considered the number of people at work in that area, and then insisted that there must be an adequate fireproof staircase within a certain distance of that central point. The size of the stairway was made to increase with the increase of that distance, which latter was not permitted to exceed a certain limit. Moreover, a stairway had to be provided not merely at one end, but at two ends at diagonal corners so that there was a possibility of reaching a stairway in two directions from a central point. The law does not permit overcrowding. Such conditions as obtain in New York City loft buildings would not be tolerated in the countries from which the people who run the industries in these lofts have come to us. Why should our communities grant them such permission here? Panic cannot be altogether guarded against in the construction of a building. Sprinklers do not constitute a full safeguard. In fact if you were to send down a douche of water upon an excitable woman when she smells smoke or sees fire you would most certainly not cool her down. Provide what safeguards you will, always keep

the crowds down to a point where even panic stricken people cannot seriously jam.

Geo. I. Rockwood. As one interested in the general subject of fire protection, I have followed closely the fire statistics of the last seven or eight years as given out by the National Fire Protection Association and the Factory Mutual Fire Insurance Companies. I have also had experience with the manner in which automatic sprinklers operate to save human life, having been responsible for the installation of many hundreds of thousands of sprinkler heads; so that with that experience back of me I want to say that no one so far as I know has really done more than has Mr. Porter to convert the rather obstinate architectural profession, as well as the equally obstinate members of the profession of fire protection engineering, to the use of this very obvious device for getting operatives and others out of a burning room and into a place of safety.

The way of the reformer is proverbially difficult, but it seems to me that those responsible for the design of loft buildings have been particularly slow in appreciating Mr. Porter's work. Very likely the reason for this will be found in Mr. Porter's scarcely concealed contempt for the automatic sprinkler as a life-saving device, but neither Mr. Porter nor anyone else has yet made any proper inquiry into the history of automatic sprinklers, considered solely from the point of view of their use as life savers. They have been regarded altogether as property savers, and are installed in buildings for that purpose. It is perfectly clear that every time a sprinkler operates successfully in a building containing human beings it is potentially a life saver. The question narrows down, therefore, to the effect of sprinklers in preventing panic conditions in buildings. Do they prevent panics or can they be made to prevent them? We need a little further light on this subject before we can dogmatize to advantage either way, but I feel confident that the solution of the problem of saving human life is a function equally of the use of Mr. Porter's centrally located incombustible partitions and the proper use of automatic sprinklers.

In the average case of a fire extinguished by automatic sprinklers three or four heads are all that open to effect the result. In the case of quick flash fires an entire roomful of sprinklers may open, seemingly at once, without any appreciable interval of time between the setting of the fire and the operation of the sprinklers. The question naturally arises, would such instant out-pouring of water be an advantage or a disadvantage to a room more or less

crowded with operatives No sane man would take the ground that he would rather be in such a room, under those conditions, without sprinklers. In a recent case where a flash fire occurred in the first floor of a three-story factory filled with people on every floor, a can of material in which gasolene was an ingredient took fire, with the curious result that, although the room was not a particularly large one, and there were many men at work in it, not a man got out of the room without getting wet. They said afterwards that there seemed to be no interval of time between the instant when the can flashed up and the subsequent out-pouring of water from all of the sprinkler heads in the room.

It might be thought that where a factory has a good deal of material out of which, say, straw hats, or lingerie, are made, the rapidity with which the fire, once started, would propagate would insure the injury of everybody in the room. Such a view would fail entirely to take into account the fact that when the fire starts the heated gases almost instantly rise to the ceiling where they proceed to mushroom out and open the sprinkler heads one after another much faster than the fire can propagate itself in the material on the floor. This is not a matter of speculation; it is an observed fact.

There are classes of buildings in which there are so many concealed spaces in the wooden walls, wooden partitions and wooden floors, that a heavy fire may conceivably get strong headway in some concealed space before it shows itself. Under such circumstances the sprinklers operate at a disadvantage and merely act as a check to the spread of the flames until the arrival of the fire department. But even then they often make it possible for the firemen to carry their hose to the very center of the fire, and thus are the primary cause of its final extinguishment.

Mr. Porter would do well to add to his crusade in favor of a central figure resistant partition, with separate means of egress on opposite sides of it, a further demand for the installation of automatic sprinklers, and he will be surprised to see how much more quickly his original effort at reform will succeed.

Harrington Emerson. In making comparative investigations of American and foreign cities what strikes one most is the much lower fire loss in German cities than in American cities. The reason is that different ideals have been pursued. Our ideal is to get there immediately. In an international test of different fire companies at Berlin about 20 years ago it took the Americans 20 seconds to come out,

couple up the hose and begin to play the water while it took the Germans over 8 minutes to prepare to fight the fire. Nevertheless the fact remains that the fire loss in Germany is small compared with what it is in this country. It is not because they have buildings that cannot be burned, because recently in Hamburg a number of incendiary fires occurred with severe losses. But the principle of the Germans is to prevent fire, not to fight it after it is started; while we have gone the limit in fighting fire after it is started. We are away behind the rest of the world in preventing fires before they start. Switzerland is a country of wooden houses yet the per capita fire loss there last year was only 2 per cent of the American per capita loss.

F. B. Gilbreth. There is no doubt that standardization more than any other one thing will reduce the number of fires; but let it be standardization for the prevention of fires rather than for extinguishing them after they are started. Fires are the product of ignorance. A very small proportion of architects, and a still smaller number of engineers, know how to construct a building that will not burn up.

In a paper on the Waste of Natural Resources by Fire,[1] Charles Whiting Baker gave a striking illustration of the annual loss not only of property but of human life. He said:

The buildings consumed, if placed on lots of 65 ft. frontage, would line both sides of a street extending from New York to Chicago. A person journeying along this street of desolation would pass in every thousand feet a ruin from which an injured person was taken. At every three-quarters of a mile in this journey he would encounter the charred remains of a human being who had been burned to death.

I have seen many of the big conflagrations in this country: Toronto, Rochester, Baltimore, San Francisco and Sioux City. A careful examination of all these ruins shows that if we tried to construct buildings that would burn up we really could not do a much better job. Reports of experiments carried on by Professor Woolson in New York City during the last ten years contain much valuable information upon this subject.

We face an entirely new situation today with the advent of concrete, which when made with the proper aggregates, such as trap rock and the right kind of sand, is of tremendous assistance in resisting the spread of fires. In fact, we are not depending on any of the old

[1]Joint meeting of the Engineering Societies on the Conservation of Natural Resources, New York, March 24, 1909.

types of construction and fireproofing schemes for preventing the spread of fires. Let us have buildings, to begin with, that will not aid the spread of fire.

I suggest for the consideration of the Society an exhibit room to which engineers, architects and insurance companies may come and satisfy themselves that there is not a single thing in construction today that cannot be made better and quite as cheaply out of absolutely incombustible material.

James B. Scott. Education, it has been said, should properly begin with one's grandparents, and undoubtedly the time to begin fighting a fire is before it begins. But granting that the "eugenics" of fire fighting have been properly observed, there will always remain the possibility of accident or incendiarism, giving rise to a hand-to-hand conflict with man's ancient enemy. The ships that can deliver more shells and heavier shells, in a given time and within a given area, and can begin delivery a few minutes earlier than the enemy, are usually awarded the victory in a modern naval engagement. The fire-fighting system which can deliver enough water, at a suitable pressure, just where it is needed, and can begin delivery in the fewest possible minutes after the discovery of a dangerous fire, is approaching the coveted 100 per cent efficiency in its limited field, as distinguished from the wider province of fire prevention.

No. 1419

PRESENT STATUS OF THE LARGE GAS ENGINE IN EUROPE

By Prof. P. Langer, Aachen, Germany

Member of the Society

The tendency to utilize in gas engines the enormous quantities of waste gases from blast furnaces had its inception in Germany about 20 years ago. The attempt was then made to create units of larger power by increasing in size the parts of smaller motors and by using a larger number of cylinders. The two-cycle system also seemed to offer a suitable method of operation for large engines, on account of its more efficient utilization of the mechanism as compared with the single-acting four-cycle system. Full success was not attained, however, until about 11 years ago when the large gas engine was brought to a high state of perfection by the Maschinenfabrik Nuernberg in the form of the double-acting four-cycle type, with two cylinders arranged in tandem. It was recognized that the principles on which to base the design of large gas engines should involve (*a*) the greatest possible accessibility of all parts exposed to the gases of combustion, and (*b*) relieving the cylinder wall of the weight of the piston.

2 These principles have not only been proven to be correct but their observance has been found to be a necessary condition for the success of large gas engines.

3 The gas engine puts much higher demands upon the attendant than any other prime mover. It creates its own potential energy by conversion of the chemically latent heat energy of the gas. Almost any defect of the machine, inherent or acquired, which interferes with the conversion of the chemical energy, acts destructively upon the machine, not to mention its influence upon the power output. The simple consideration that gas is being burnt without rendering its equivalent in power, leads to the conclusion that the balance is being transmitted to the cooling water under pressures and temperatures beyond what are permissible, and that exhaust gases unallowably hot

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

are flowing around the exhaust valves. In the case of the steam engine a leaky piston does no further damage than to increase the steam consumption. The governor automatically adjusts for a longer cut-off and the engine pulls through as long as the boiler furnishes sufficient steam. With the gas engine, however, a leaky piston causes ignition at the wrong time which means a release of heat energy at a time and place when this energy can do no useful work. A continuation of the operation under such conditions is not permissible. Any attempt to force the operation would cause heavy damage to the machine.

4 It is the duty of the designer to face these facts and to design the machine in such a manner that it is possible to give it the careful attention needed to prevent such disturbances. Such attention cannot be given when access to such vital parts as pistons, stuffing box packings and valves can be had only after a long and difficult job of disassembling. The author knows of an engine which, after six years of continuous day and night operation, showed only 0.015 in. wear in the diameter of the cylinder. This success, although immediately traceable to the careful attention this machine received, is nevertheless indirectly due to the designer who made it possible to give such attention through the excellence of his design.

5 An instance of defective design in this respect is the case of a stuffing box in which it is possible to replace the packing rings only by unscrewing the joint between the piston rod and crosshead. The packing rings must be split and the whole packing must be easily removable in order that a new packing may be put in during a brief interruption of the operation. The result of neglecting this seemingly minor point of design is a continuation of operation with leaky packings and danger of warping the piston rods.

6 A further basic condition for uninterrupted operation is pure materials, pure gas and pure cooling water.

7 Operation with impure materials puts demands upon the operator which it is impossible to meet, except by considerable reduction of the output on account of the long interruptions required for cleaning. It is evident that the economy of the plant suffers materially through reduction of the output, as the proportion of the unproductive capital, the idle machinery, increases.

8 The attempt to attain stratification of the mixture inside of the cylinder has led to very complicated valve gears. It was hoped that a gas valve, which remained open during only a part of the suction stroke would direct the gas in such a manner that a combustible mixture would be present at the point of ignition even under the lightest loads, while the balance of the combustion space would be filled with inert air. The result, however, did not justify this hope. Instead of stratification there was only a bad mixture resulting in irregular and uneven operation. Today the hope of attaining stratification can be considered as being finally disposed of and designers have returned to the simple throttling valve gear.

9 Throttling of gas and air simultaneously, or in other words, regulation of the quantity of mixture only, is to be preferred to the throttling of gas only, as the former method makes possible a more certain control of the power developed.

10 Quantity regulation will give satisfaction only if the action of the governor upon the throttle valve has been given careful consideration. Present and older designs that are faulty in this respect are frequently seen. It is wrong to let the governor act in such a way that the valve opening is proportional to the effective travel of the governor sleeve, as the following consideration will show:

11 The result of throttling is to reduce the quantity of mixture drawn into the cylinder during each stroke, on account of the reduced openings for gas and air. The volume of the charge remains the same, as the cylinder is always completely filled. The density of the charge will become less and correspondingly its weight and the amount of energy supplied.

12 These relations are shown in the curves of Fig. 1. Starting with a certain velocity C of the mixture in the valve, which is determined by the opening of the valve, the curve shows the velocity as a function of the position of the governor. This velocity can be produced only by a certain drop in pressure as the mixture passes to the cylinder, so that the law according to which the absolute pressure in the cylinder changes is also definitely determined.

13 The density of the charge is proportional to the absolute pressure and the absolute weight of the charge of constant volume is proportional to the density, and so is the amount of heat energy supplied. The shape of this curve shows that the lower part of the

travel of the governor sleeve is almost without influence and that the total regulation is limited to the upper travel.

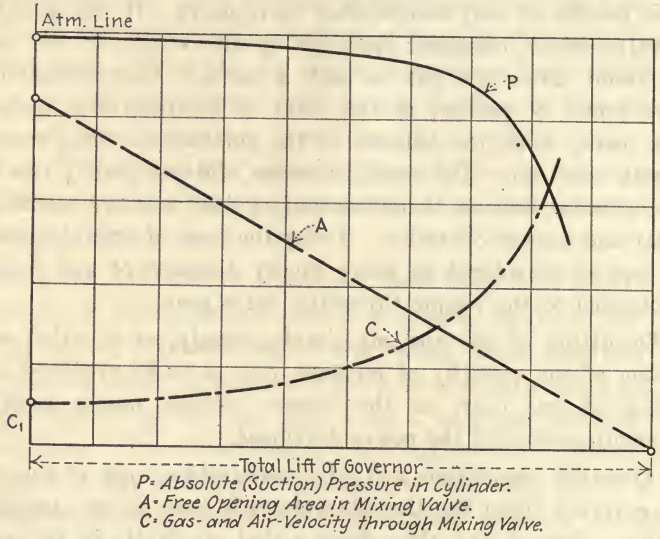

FIG. 1 CURVE SHOWING RELATIONS OF OPENING OF MIXING VALVE, UNDER CONTROL OF GOVERNOR, TO SUCTION IN CYLINDER AND VELOCITY THROUGH VALVE

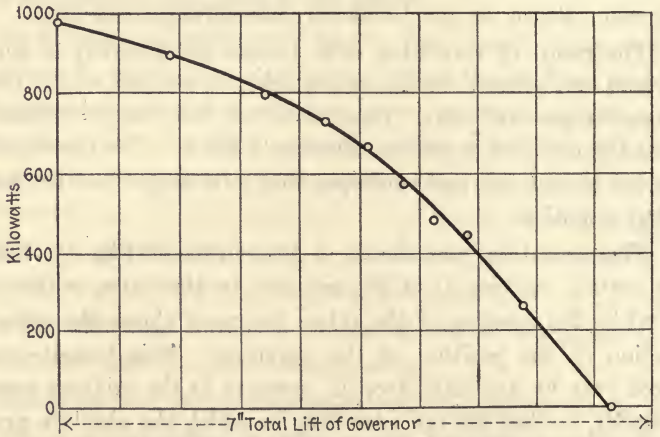

FIG. 2 CURVE SHOWING EFFECT OF PROPER ADJUSTMENT OF GOVERNOR MECHANISM TO OBTAIN UNIFORM UTILIZATION OF TOTAL LIFT OF VALVE

14 The result of this wrong regulation is marked irregularity of the indicator diagrams, as the smallest motion of the governor causes considerable changes in the quantity of mixture supplied. Combustion

is irregular, there is tendency to backfire, or in one word, the machine "does not govern."

15 The remedy is found in changing the connection between the governor sleeve and the throttling mechanism in a manner such that in the low positions of the governor the throttling action is more intensive than in the upper positions. This action can be accomplished in a simple manner by off-setting the connecting link between the governor and the throttling valve, similar to the arrangement found in Corliss valve gears. This allows a uniform utilization of the effective lift of the governor, and consequently stable and quiet regulation, without any complication whatever. Fig. 2 shows a diagram, characterizing

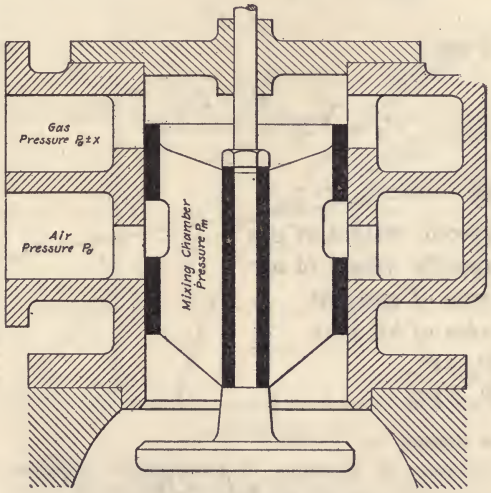

FIG. 3 TYPICAL DESIGN OF MIXING VALVE FOR LARGE GAS ENGINES

the regulation as taken from an engine having a properly designed throttling gear.

16 Even the best considered scheme of regulation will not avail, if the proportion of mixture, that is the ratio of gas to air, is not properly controlled. In engines that have separate pumps for air and gas, as usual in two-cycle machines, where therefore, air and gas are furnished in measured quantities, it is a comparatively easy matter to get the proper proportion, at least at full load. Difficulties are found, however, in four-cycle machines, where gas and air are drawn in in parallel by the working piston. In this case the proportion of mixture is not readily controlled, because the quantities drawn in depend not only on the free opening of the admission valve,

but on the product of this opening area and the velocity of flow through it. Incidental changes in pressure, due to static or dynamic causes, may influence this velocity strongly.

17 If we denote by P_g and P_a the pressure in the gas and air pipes respectively, before they reach the mixing chamber (Fig. 3), and by P_m the pressure prevailing at the same time in the mixing chamber, then P_g—P_m is the difference in pressure creating the velocity of gas and P_a—P_m the difference in pressure creating the velocity of air. Consequently velocity of gas

$$C_g = \sqrt{2g \frac{P_g - P_m}{\gamma_g}}$$

and velocity of air

$$C_a = \sqrt{2g \frac{P_a - P_m}{\gamma_a}}$$

If we denote by

γ_g = specific weight of gas
γ_a = specific weight of air
A_g = area of gas-port
A_a = area of air-port
$H_g = P_g - P_m$
$H_a = P_a - P_m$

Volume of gas drawn in

$$V_g = A_g \sqrt{2g \frac{H_g}{\gamma_g}}$$

Volume of air drawn in

$$V_a = A_a \sqrt{2g \frac{H_a}{\gamma_a}}$$

Therefore the ratio of mixture

$$M = \frac{V_g}{V_a} = \frac{A_g}{A_a} \sqrt{\frac{\gamma_a}{\gamma_g} \times \frac{H_g}{H_a}}$$

18 It may be noted that the difference in specific gravity must be considered when deciding upon the proportions of the ports for correct mixture in engines that use gases lighter than air. A disregard of

this fact leads to too rich mixtures and the engine becomes choked with gas. The many failures of engines working with coke oven gas are partly explained by wrongly proportioned gas ports. At least, it was found that proper operation resulted in many cases from materially reduced gas ports.

19 If, for the sake of simplicity, we assume in the case of blast furnace gas, the ratio of the port areas as well as the ratio of specific gravities, to equal unity; and if we assume the equation $H_g = H_a \pm x$, where x denotes the difference in gas pressure, as compared with the

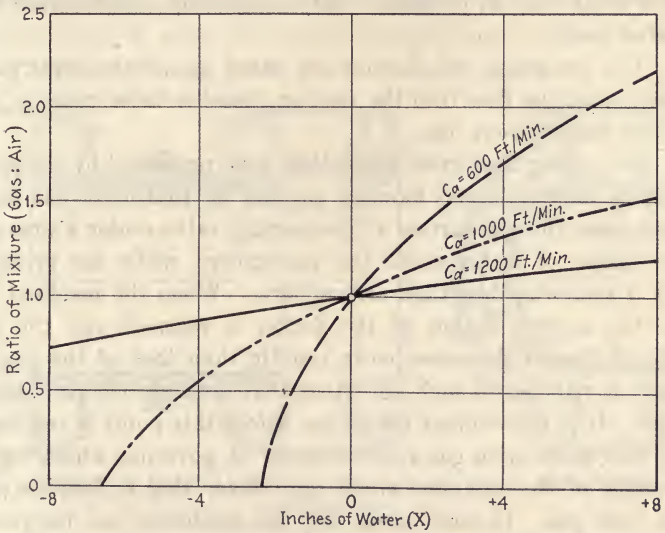

FIG. 4 CURVE SHOWING VARIATIONS IN RATIO OF MIXTURE OF GAS AND AIR WITH DIFFERING VELOCITIES OF AIR DRAWN IN

air pressure, then we have the very simple relation, for the ratio of mixture

$$M = \sqrt{1 \pm \frac{x}{H_a}}$$

This equation shows what means have to be employed in order to keep the variations of the mixture within permissible limits. It shows the necessity of keeping H_a high, which means that we must work with high velocities in the mixing valve, because the larger the value of H_a, the smaller the value of $\frac{x}{H_a}$ for a given value of x, and the smaller the variation of M from the desired value 1.

20 Fig. 4 shows the curves of M for different velocities of intake (600, 1000 and 1600 ft. per min.). These curves show that the intake velocity in the mixing valve must be at least 1000 ft. per min. in order to avoid disturbing variations of the mixture due to accidental changes of the gas pressure. Practical experience fully confirms the correctness of this consideration. Velocities of about 1200 ft. per min. must be considered as normal. In order to avoid too intensive a throttling action and consequent reduction of power, the piping and other passages leading to the mixing valve must be proportioned liberally, so that the total drop in pressure is advantageously concentrated within the valve itself.

21 The foregoing calculations are based upon the simplifying assumption that the flow into the mixing chamber takes place at a uniform rate and without loss.

22 For a long time great difficulties were presented by the problem of driving variable speed blowing engines by four-cycle gas engines. In most cases the gas arrives at the mixing valve under a pressure of several inches of water above the atmosphere, while the pressure of the air is somewhat less than atmospheric. When the machine is run slower the suction action of the piston is reduced and the intake velocity of the air decreases more rapidly than that of the gas, until finally it is reduced to such an extent that nothing but gas enters the cylinder. It is self-evident that, even before this point is reached, the engine will choke with gas and will stop. A governor which regulates the quality of the mixture would only favor this suffocation of the engine with gas. In such cases, too, the conditions are improved by intensive throttling at normal speed.

CYLINDERS

23 Improvements of design as well as improvements in foundry practice have reduced the breakage of cylinders to a point where it is now rather a rare occurrence. Views upon the most suitable shape still differ, however, considerably. While in all other details of large gas engines standard designs have been developed, which serve all purposes, the cylinder designs still vary considerably. The fact that some designers, after careful experiments, have abandoned the split jacket cylinder in favor of the normal one-piece cylinder, while others, also after careful experiments, have gone in exactly the opposite direction, and split not only the jacket, but also the inside cylinder, Fig. 5, seems to show an uncertainty in judging the causes of breakage. These differences in design, however, are caused largely by fixed ideas

of the purchasers whose special wishes are complied with by clever salesmen. The one-piece cylinder is just as strong as the split one. The split cylinder has come principally from the desire to avoid initial stresses in the direction of its axis, which put the inner cylinder under tension on account of the fact that in casting it cools later than the rest of the casting. Besides, by casting the two halves separately, it was attempted to obtain as close-grained a wall as possible for the combustion chamber. Finally, splitting the cylinder has the advantage that customers who consider a cylinder liner the proper construction can be satisfied, inasmuch as the insertion of liners in one-piece cylinders presents some difficulties, though these are not to be considered insurmountable.

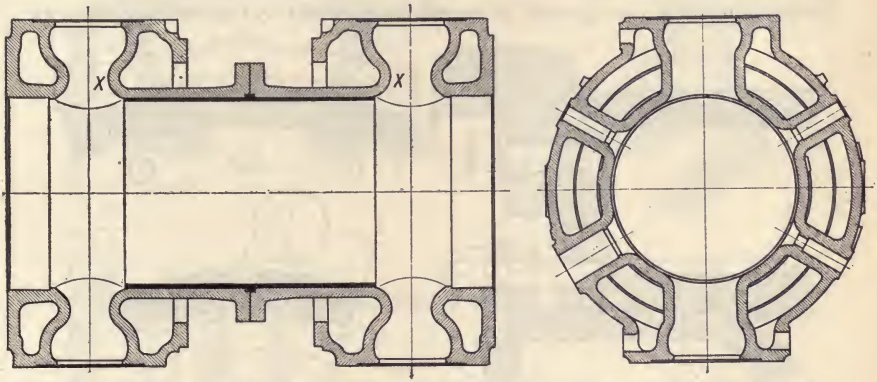

FIG. 5 DESIGN OF SPLIT-JACKET GAS ENGINE CYLINDER WITH SPLIT INSIDE CYLINDER

24 Besides the advantage of simplicity and the absence of the danger of leakage in the highly strained cylinder joint, the one-piece cylinder allows a better transmission of the forces from the cylinder to the frame. The fact that one-piece cylinders frequently broke at the place indicated in Fig. 6, caused an over-estimation of the stresses in the casting. Closer investigation shows that the breaks are caused simply by making the part between cylinder and flange too weak. The wall, besides being too weak originally, is further weakened by the large number of cover studs and the break is further induced by excessive tightening of these studs. This fault can be remedied by properly reinforcing the point of danger and by placing the joint shoulder on the outside. As a matter of fact, breakage on account of strains in the casting or from expansion due to heat, has not occurred in cylinders that were properly reinforced. Improvements in foundry practice have undoubtedly helped to avoid such breakage.

25 While the breaks just discussed have their origin in the strains set up by irregular cooling of the casting or by irregular heating in operation, cracks of an entirely different nature have been observed on the walls of the combustion chamber. These so-called fire cracks always start in a place where the transmission of heat to the cooling water was impeded by some cause or other (see Fig. 7). The cause of these cracks is to be found in the stresses produced by unequal temperatures within the same wall. In the smooth and homogenous wall the differences in temperature of the different strata are small and the temperature of the wall is comparatively low, as long as accumulated scale or similar causes do not offer an obstacle to the transmission of heat. When, however, the capacity of the wall for conducting heat, or the rate of transmitting heat to the cooling water is

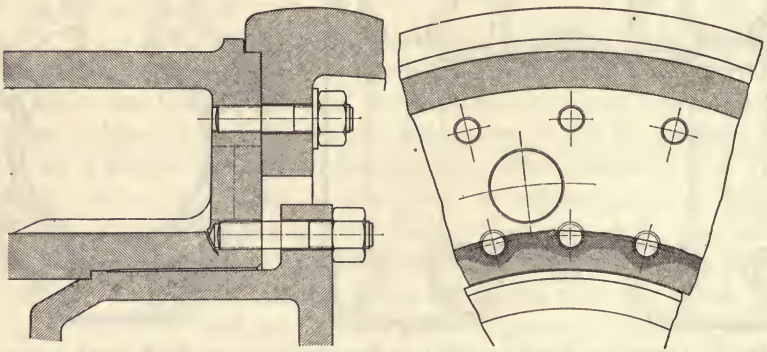

FIG. 6 DETAIL OF POINT OF FREQUENT BREAKAGE OF ONE-PIECE GAS ENGINE CYLINDERS

reduced, there will be an accumulation of heat which will cause a considerable rise of temperature during the expansion and exhaust strokes. The material will tend to expand according to the average temperature of the wall and it will be able to do this without resistance because it was under tension on account of the strain in the casting. In operation, therefore, the strain will be relieved. On account of temperature differences, however, in different strata of the wall, there will be stresses in the wall itself, compression in the hotter zones and tension in the cooler ones.

26 This condition of stress can be compared with that existing in a bar being bent towards the inside of the cylinder. In the hotter layer the expansion being resisted causes compression stresses, and in the colder layers there will be tension stresses. As soon as the cold mixture is admitted during the next suction stroke, the surface of the

wall is cooled intensively. The inside layers of the wall cannot follow rapidly enough to cause the establishment of settled conditions corresponding to this flow of heat. The mean temperature of the wall, and consequently its average expansion can only be affected very slightly during the first moment of internal cooling by the entering cold mixture. The innermost layer will be under strong tension on account of the sudden cooling. This "jumping" of the temperature acts upon the material in much the same manner as sudden flexure from the inside to the outside would. The inside layer of the wall of the combustion chamber, therefore, is exposed to the same kind of stresses as

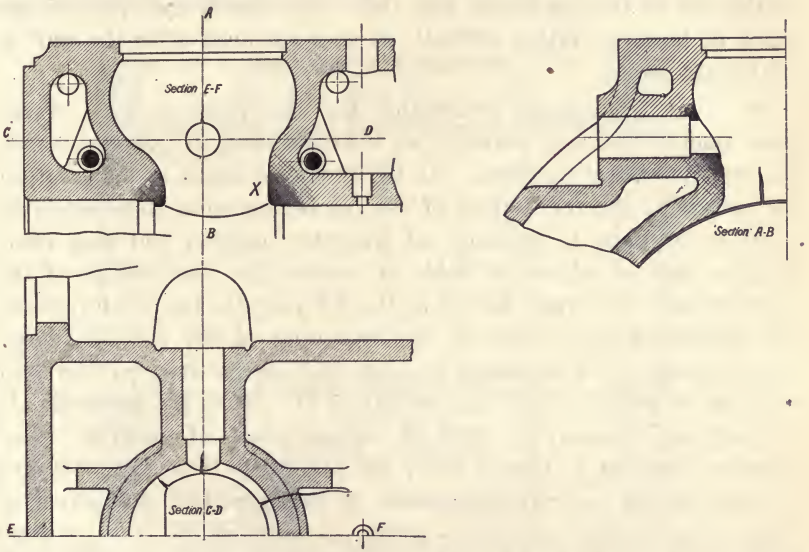

FIG. 7 TYPICAL LOCATIONS OF FIRE CRACKS IN PLACES WHERE COOLING IS IMPEDED

the outside fiber of a bar that is continuously bent in both directions by blows. The stresses which occur under these conditions are about proportional to the difference in temperature, the coefficient of heat expansion, and the modulus of elasticity, if, indeed, one can speak of such in the case of cast-iron.

27 This is an extraordinarily unfavorable case of strain. Cases of strain on account of sudden changes of temperature are not rare, but these conditions are worst in the cylinders of gas engines, on account of the rapid succession of the changes in temperature. A mathematical deduction of these stresses is out of the question on account of the impossibility of obtaining even halfway accurate data on the distri-

bution of temperature at any one moment over the different zones of the wall.

28 If we bear in mind, however, that a difference in temperature of 200 deg. fahr., when the expansion is restrained, corresponds to a strain of about 15,000 lb. per sq. in., we are surprised to find that cracks do not occur more often. The occurrence of a crack, which in the beginning is scarcely ⅛ in. deep, apparently relieves the strain to a certain extent and only the natural tendency of the cast material to continue to break together with the external mechanical forces causes the crack to open further. In most cases drilling and calking at the end of the crack will stop this. The timely discovery of the crack is, however, rather difficult, as it is not open after the wall is uniformly cooled.

29 The first step in combatting the occurrence of these cracks must reduce their real source, i. e., the differences in temperature, to an amount that is harmless. As the sudden changes of temperature are caused by the very nature of the gas engine cycle, these attempts must be confined to avoiding all irregular ignition and slow combustion, both of which are liable to increase the temperature of the cycle beyond the normal as well as the difference in temperature when the sudden change occurs at the beginning of the suction stroke. Furthermore, it is necessary to make the conductivity of the wall uniform in order to avoid accumulations of heat in the material. It is therefore necessary to avoid all accumulations of material. Passages as shown at *X*, Figs. 5 and 7, are also bad. Here accumulations of heat are the natural consequence of the imperfect conduction of heat being directed towards a center, the passage at the same time being very much exposed to the cooling action of the incoming air. Places like these are predestined to suffer from heat cracks.

30 The commonly used globe or onion-shaped passages for inlet and exhaust valves are therefore not suitable, and lately the form shown in Fig. 8 is very properly preferred. Here the valves are brought close to the inner surface of the cylinder.

31 In a much more effective manner than by measures of design, can the heat cracks, and in fact all cracks that occur in gas engine cylinders, be avoided by proper choice of a material. The constant of this material (coefficient of heat expansion multiplied by modulus of elasticity), must be less than is the case with cast iron. The less the expansion from heat, which is the real cause of the strain, and the more elastic the material, the less the strain.

32 Considering the enormous progress which metallurgical science has recorded during the past few years, a solution of this question of material should appear possible, and the more so, as nickel steel alloys have actually been made for accurate rules, in which expansion from heat cannot be detected at all.

33 An investigation of the constant in question (coefficient of heat expansion multiplied by modulus of elasticity) upon which depends, according to the foregoing, the strain of the material resulting from uneven temperature, shows that cast steel is not a suitable material for gas engine cylinders. The larger expansion due to heat as compared with cast-iron and the very much higher modulus of

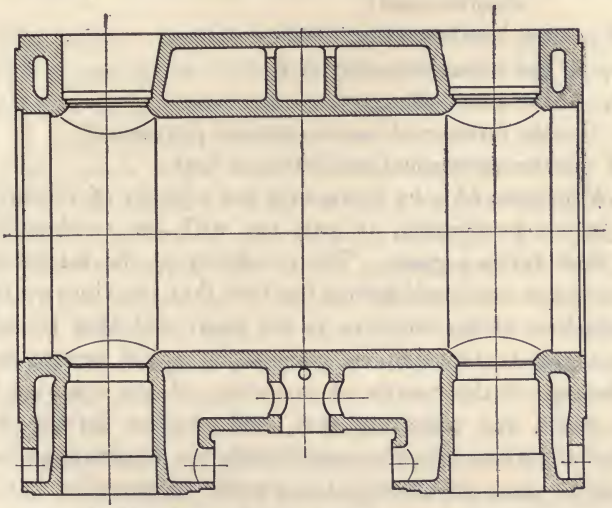

FIG. 8 RECENT DESIGN OF ONE-PIECE GAS ENGINE CYLINDER THAT AVOIDS GLOBE-SHAPED PASSAGES TO VALVES

elasticity causes increased stresses, while the strength of cast steel is not proportionately greater. Experience confirms this conclusion.

ON INCREASING THE OUTPUT OF LARGE GAS ENGINES

34 The principal task upon which the gas engineer is laboring incessantly, besides that of increasing the reliability of the machine, is that of making it cheaper, partly by simplifying its construction, partly by increasing its output. At present it can hardly be imagined that the gas engine can be made cheaper by making its construction simpler, now that the complicated valve gears have disappeared. Reduction of weight is not to be recommended. Until recently experience has demanded a continuous increase of engine weights. Economy

in material would reduce reliability and length of life of the machine. The mechanism has been developed into standard designs, so that in this detail there is nothing to be saved.

35 Somewhat better prospects seem to be opened by the attempt to reduce the cost by increasing the output of the machine of a given size. Let us consider the equation for the power of the machine

$$W = \frac{1}{A} H V \gamma \eta_v \eta_t$$

in which

V = the swept volume (the factor which determines the absolute cost)
H = the heating value of the mixture
η_v = the volumetric efficiency
η_t = the total efficiency
γ = the number of suction strokes per second
A = the mechanical equivalent of heat

36 An increase of γ by increasing the number of revolutions per minute, is not practicable, at least not with less combustible gases such as blast furnace gases. The reliability of the machines would be lessened, not even considering the fact that the time available for the combustion of the mixture is too short and that therefore the losses on account of incomplete combustion would become too large. A comparison of the results of operation of gas operated blowing engines, which run relatively slow, and engines driving dynamos, proves that the former require considerably less repairs than the latter. Increasing the number of revolutions would increase the idle time on account of repairs, and a gain in output can hardly be expected.

37 The heating value of the mixture H, is also determined by the minimum excess of air necessary for complete combustion, and there remain only the two factors of volumetric and total efficiency. It is conceivable that the total efficiency might be increased by increasing the thermal efficiency of the process. This, however, is dependent upon increase of the pressure and for this reason such a possibility must be eliminated. The volumetric efficiency can be increased in two ways; by cooling the charge and by increasing the pressure of the charge. The first method was proposed some time ago. It is not being adopted because in most cases the cooling water is warmer than the atmosphere and the incoming gas. Besides, a large gain cannot be expected, as decreasing the temperature of the charge 10 deg. fahr. could only increase the output about 2 per cent.

38 The other method for increasing the output, which lately has attracted renewed interest, consists in increasing the pressure of the charge and simultaneous scavenging of the exhaust gases. In this manner a given suction volume of an engine can accommodate a larger weight of charge and greater output of power can be attained.

39 The original purpose of this method, which has been known in England for many years, was the avoidance of pre-ignition by scavenging the cylinder of what remained of the hot exhaust gases. It is the competition into which the gas engine has lately had to enter with the steam turbine, that has brought to the front the possibility of increased power output and consequently lower cost of the machine per unit of power.

40 The simplest method of realizing the scheme of scavenging and charging consists in closing the exhaust valve late and opening the inlet valve early. The time during which both valves are open is used for scavenging. In order surely to avoid loss of gas, the charging with gas is only commenced after the exhaust valve is closed. At the end of the suction stroke the mixing chamber must be cleansed of combustible mixture by scavenging (rinsing) with pure air, as otherwise backfires would occur at the beginning of the next scavenging period. Gas and air are brought to the machine under a pressure of about 3 lb. per sq. in. gage. On account of this increased pressure the weight of the charge is about 20 per cent greater than that of the normal method. Scavenging the combustion chamber also furnishes further space for fresh mixture. Experiments have shown a mean effective pressure of about 100 lb. per sq. in. In continuous operation, in so far as we can speak of such in experiments, mean effective pressures of 85 lb. per sq. in. can be reached.

41 Undoubtedly such results present much that is attractive, especially as the output can still be increased by further increasing the pressure, and as this would give to the gas engine the capacity for overloads which heretofore it did not possess in the same sense as steam engines do. Experiments have also proved that these overloads can be had without an increase of the maximum pressure. This is done by reducing the compression.

42 In spite of the lower compression the heat consumption per unit of power was not increased. It seems that this result is explained first by better combustion of the charge which is not contaminated by the remains of the exhaust gases, and secondly by a decreased percentage of loss in the cooling water. The predominating influence can only be found by means of carefully measured heat balances which at present

are not available. The increased temperature of the cylinder walls, which has been found thermo-electrically, has caused objections against continued operation with scavenging and forced charge. A certain justification cannot be denied to these objections. The *decrease* of continuous loads is the main factor that has contributed to overcome the difficulties of operation. To endanger regular operation by increasing the mean effective pressure must appear to be a dangerous experiment in just such cases where the equipment is insufficient and an increase is highly desirable. The disposition on the part of operating engineers to await developments, is easy to understand. The present satisfactory operation, which is the result of years of experience, is making these engineers justly conservative.

43 The danger is to be found in the fact that the increase of output on one hand is offset by a decrease on the other hand on account of the more frequent shutdowns for repairs. Besides these questions, which can only be solved by several years of experience, there are other difficulties in the constructive realization of this idea of increased output which have to be solved before it is proved to be practicable in sustained operation. In one word, there are many nuts to be cracked, before we can think of a practicable increase of the power output of four-cycle engines by means of scavenging and forced charge.

IMPROVING GAS ENGINE HEAT ECONOMY

44 The endless desire of gas engine builders to improve the economy of the gas engine, has lately brought to the front interest in the utilization of waste heat. According to experience, a blast furnace gas engine dynamo consumes in continuous service on the average about 16,000 B.t.u. per kw-hr. Of this amount of heat only 3412 B.t.u. are converted into electrical energy. Of the balance, 12,588 B.t.u., only the mechanical, electrical, and radiation losses have to be regarded as unredeemable, as well as the energy contained in the unburned gases. The heat in the cooling water, about 4800 B.t.u., and that in the exhaust, about 5200 B.t.u. per kw-hr., is, however, available for a more or less perfect further utilization. The high temperature of the exhaust, about 880 deg. fahr., measured at the exhaust flange, makes possible its immediate use for the generation of steam in boilers, which have to be placed as closely as possible to the exhaust chamber. As a matter of fact, it is possible to obtain about 2 lb. of high-pressure steam per kw-hr. of the gas engine.

45 The utilization of the heat in the cooling water is not possible as directly as this. The further use of this heat for power purposes means the generation of steam and this requires a considerably higher temperature of the cooling water than that customary today in large gas engines. Semmler suggested as long ago as ten years, that gas engine cylinders be cooled with water hotter than 212 deg. fahr., and that this water be put under pressure in order to avoid the generation of steam in the jacket. Evaporation commences only outside of the jacket, when the superheated water is conducted through a throttle valve into a steam drum. It may be easily understood that this "hot cooling" scheme has been regarded rather skeptically for a long time. About a year ago it was decided at the Rombacher Huetten Werke to try Semmler's cooling method on an 800-kw. tandem engine. The results in operation, which are available today, are entirely favorable. The hot jacket not only did not hurt the cylinder, but it caused apparently a softer and more quiet operation of the machine.

46 The use of distilled water in a closed cycle like Semmler's, the considerably higher temperature of the outer cylinder wall, which causes nearly equal average temperatures of the inner and outer cylinder walls, are surely advantages which, if the favorable experiences last, merit more consideration than the recovery of waste heat. The amount produced is in the case of the engine mentioned, about 0.5 kg. of steam of low pressure per kw-hr. There is no doubt that this figure can be very much improved by designing the cylinders more suitably for this purpose, by careful insulation and by connecting the cooling water from the piston with this system, which heretofore has not been done.

47 The results achieved in recovery of waste heat will be of different practical value for different types of blast furnace plants. Where there is a steam plant in addition to gas plants, it would be advantageous to shut down some of the boilers and substitute steam from the waste heat boilers by connecting them to the steam mains, because this steam can be produced almost without cost for attendance and for fuel, and therefore, as cheaply as is possible.

GAS ENGINE VS. STEAM TURBINE

48 In conclusion it may be well to touch in a few words, upon the competition between the gas engine and the steam turbine, which lately has become very keen in Germany as well as in the United States. The steam turbine has doubtless made much progress

during the last few years. In its perfected form in which we know it today, it is a thoroughly reliable machine. Its overload capacity, with nearly constant steam consumption per power unit, is limited by its accessories, as boiler and condenser. In addition, there is its principal advantage of lower first cost as compared with the gas engine. Its disadvantage is the higher consumption of heat per unit of power. The gas engine produces approximately twice the amount of energy from a given amount of heat, and is therefore, superior as a fuel saver from the economic point of view. To the business man, however, the question of lower first cost often appeals more strongly. The favorable balance sheet of the current year pleases him better than the quiet satisfaction in the knowledge that he has saved valuable treasures of fuel for his great-grandchildren. But even from this standpoint the gas engine will be victorious over the steam turbine in cases where the available waste gases are not so abundant that their combustion under steam boilers would suffice to satisfy the existing requirements for electrical energy. Whether and to what extent these conditions will be changed by the new method of surface combustion, is a matter which the future will show.

DISCUSSION

H. J. K. Freyn. Professor Langer devotes a considerable portion of his paper to the question of regulation of large gas engines and he points out in a very able and clear manner that of the two prevalent systems of regulation of gas engines, namely: by stratification of the mixture and constant compression versus throttling of gas and air and variable compression, the latter method has proved its superiority in every instance and from every point of view.

I can but heartily endorse every statement made by Professor Langer and while in my earlier career I favored the stratification method, I changed my mind many years ago as a result of practical experience with engines regulated on both systems.

Abroad, the leading gas engine manufacturers have for a considerable length of time adhered to the stratification principle, although choosing the more rational form of attempting to obtain stratification by arranging their valve gear in such a manner that a certain quantity of pure air always *followed* the mixture.

In this system, therefore, only one chance for diffusion between gas mixture and pure air exists, with the result that operation at fractional load was reasonably satisfactory because the danger of

formation of a bad mixture during the suction and compression strokes was not as great as it is in the system where a variable quantity of mixture is "sandwiched" between two layers of pure air, causing diffusion in two planes, as it were, on either side of the mixture.

It has been my experience that gas engines operating on the so-called stratification principle, while giving excellent results from full load to approximately half load, are not capable of maintaining regular ignition upon all piston faces as soon as the load drops below approximately 50 per cent of the rated capacity.

The influence of this phenomenon upon regularity of operation, especially if alternating-current generators driven by such engines have to be operated in parallel, is very marked and it will be found that such power plants usually show excessive cross currents and "swinging on the line" at light load. This is not the case with engines regulated on the principle of variable compression and constant mixture, and especially with those where gas and air are throttled during the whole suction stroke.

Professor Langer has very ably and comprehensively proved along theoretical lines why this should be so, and he points out especially what provision must be made in the governor gearing to bring about the desired results, not only at full load but also at fractional load.

The point of "drowning" of gas engines in gas, raised by the author, is very well taken. Gas blowing engines which have to be operated at times at reduced speed are particularly susceptible to drowning. Gas blowing engines regulated on the principle of constant mixture and variable compression are better suited for slow speed operation, as was proven in many instances in Europe where large gas blowing engines could be operated for one hour at a speed of 19 r.p.m. with full pressure of 15 lb. on the blowing cylinder.

Without a single exception, German gas engine manufacturers several years ago abandoned the stratification method of regulation and a large majority have adopted the so-called "combination system," making use of the simplest method imaginable for throttling of air and gas, viz., by the application of so-called butterfly-valves as throttling and regulating organs.

These engines are regulated in the following manner. From full load down, the governor acts first only upon the gas dampers with decreasing load, while the air dampers remain in their original position. Below about 60 to 70 per cent of rated load, the governor begins, through a toggle motion, to act upon the air butterflies as

well. The toggle effect is so arranged that with further decreasing load, the air dampers act more quickly than the gas dampers, so that below a certain fractional load a perfectly constant mixture is admitted.

It will be seen that this combination method of regulation combines the advantages of both stratification and constant mixture regulating methods; at the heavier loads, practically constant compression is obtained, while at the lighter loads governing is performed on the constant mixture principle.

With reference to the subject of gas cylinders, I believe that Professor Langer's excellent exposition of the relative merits of cast iron and cast steel and of the one-piece vs. the split gas cylinder deserves the careful attention of both gas engine builders and users. Professor Langer's discussion of this question is very timely, because in this country the battle between the advocates of the one and those in favor of the other is still waging.

I have had an opportunity of studying these questions in detail during the last few years and I have come to the conclusion that Professor Langer is unquestionably right in his preference for one-piece cast-iron gas cylinders fitted with hard cast-iron liners. Earlier difficulties with cracking of gas cylinders have naturally led manufacturers to look for a building material which could stand up better under the strains and stresses imposed by mechanical forces and temperature variations. In Europe cast-steel cylinders were tried several years ago with the result that breaks occurred after a much shorter time of operation than when cast-iron cylinders were used. I believe that a similar experience was had in this country.

Cast-iron cylinders in one piece can be made so perfect today that they will stand up very well. At the gas engine power plant at Gary, 140 one-piece cast-iron gas cylinders are in operation, 68 of which have been in service nearly five years. Of all these cylinders only one had to be actually replaced by a new one, although at the time of installation of these particular engines, the art had not progressed as far as it has today. It is true that a number of these cylinders show fire cracks and breaks in the counterbore, but repairs have been made and the cylinders are still in perfectly satisfactory operation.

One of the most important points to which by far too little attention is paid, especially in this country, is the question of the use of suitable cooling water. Very interesting experiments were made abroad to determine the effect of the accumulation of scale on the

cylinder walls and it was found that the average temperature of these walls increased at an amazing rate with seemingly unimportant deposits of scale. I know of at least one installation where thermometers are inserted permanently in the inner cylinder walls reaching to within ¼ in. of the cylinder bore. The operator is supposed to watch these thermometers which, after the cylinder jackets have been thoroughly cleaned with wire brushes and weak acid solutions, show a gradual rise of temperature while operation continues. As soon as the thermometers show a temperature of 125 deg. cent. the engine is shut down and the cylinder walls are again carefully scrubbed and cleaned. In this particular instance the cracking and breaking of gas cylinders which had become a nuisance on account of its frequency, was entirely stopped.

Professor Langer has elaborated considerably on the question of the increase of output of large gas engines and improvements in gas engine economy. I have studied the latest attempts of European gas engine manufacturers to obtain an overload capacity of four-cycle gas engines by using the so-called scavenging and surcharging method. I have seen several installations abroad equipped with this system and I am familiar with tests made by Ehrhardt & Sehmer on engines furnished by them. They show that in spite of a material increase in mean effective pressure amounting to 25 to 35 per cent, no increase occurs in the initial pressure nor in the average temperature of the gas cylinders and other gas engine parts exposed to the high temperatures of combustion. This firm is prepared to give guarantees regarding the heat consumption of engines equipped with the scavenging and surcharging system which are not any lower than those usually given for ordinary four-cycle engines of equal capacity.

I cannot agree with Professor Langer's statement that engines operated on this principle are subject to a greater wear and tear than ordinary gas engines; the stresses and strains which the running gear and other engine parts sustain are due primarily to the initial pressure and not to the mean effective pressure. If we consider the main bearings, for instance, it will easily be seen that such a bearing will give no trouble as long as a film of oil can be maintained between bearing shell and shaft. The maintenance of such a film depends upon the amount of pressure per square inch and not upon its duration; with lower initial pressure and higher mean effective pressure, therefore, these main bearings must give at least as good satisfaction as they are giving in ordinary four-cycle engines. As a matter of

fact, the dimensions of pins, bearings etc., of scavenged and surcharged engines have not been increased beyond those customary in ordinary non-scavenged engines.

The question of utilization of the exhaust heat of gas engines has had practically no attention in this country, and yet it is such an excellent and cheap means of increasing the usefulness of the gas engine and the financial returns from its application. I wish to refer in this connection to one of the best papers on the subject, which was read by Leon Greiner before the Liege Engineering Society about a year ago. Practical results are given which were obtained with an installation of such boilers in connection with blast furnace gas engines at the John Cockerill Works of Seraing, Belgium.

Referring to the competition between gas engines and steam turbines, I wish to call attention to my paper read before the American Iron and Steel Institute's meeting of last May, which is devoted exclusively to this subject and which proves with the aid of cost figures obtained in actual, commercial operation of large gas engine power stations in this country that gas engines operated on so-called "waste" industrial gases—an unfortunate misnomer which should be abolished as quickly as possible—are superior in commercial economy to any other known method of producing power.

F. Z. Nedden said that the present status of the large gas engine, both in Europe and in this country, is perhaps more considerably influenced by the introduction of surface-combustion boilers than implied in the paper. In this country we have very largely the use of natural gas, especially in the Pittsburgh district, which is very suitable for being used under Schnabel-Bone surface combustion boilers. One of these installed at the Shinningroove Iron Works, in England, is claimed to have given an efficiency of nearly 90 per cent, as compared with the efficiency of the ordinary boiler, which ranges from 65 to 70 per cent. That would mean that with the surface-combustion boiler the gases available in the Pittsburgh district would be able to produce one-third more power than at present. In other words, the cost of energy would in that district be reduced by about 20 per cent.

A great deal is said about the advantages of the Schnabel-Bone boiler which are naturally claimed by those interested in its introduction. Professor Langer's experiences with it would be of interest, if he would give some account of them.

To make his questions more specific, he said he would ask Professor Langer if it was true that the efficiency of the Schnabel-Bone boilers actually exceeded that of the ordinary boiler by about 25 per cent, and if that boiler was reliable in service[1]; also if the utilization of the heat of waste gases in the gas engine was possible by the Schnabel-Bone boiler?

R. H. FERNALD. At a central plant in this country, built to utilize low-grade fuels at the mines, I ventured to criticise the fact that a steam plant had been constructed instead of a gas plant. The vice-president of the company said that he had wanted to put in a gas plant, but was forced to put in steam as he found no large gas producer units that would be available and satisfactory for an installation of the size contemplated. In other words, it seems to be a question of the size of the unit which is deferring the installation of plants of 100,000 to 250,000 h.p. directly at the mines. I would, therefore, like to hear from Professor Langer his opinion regarding the possible size of the large gas engine of the near future. The question, of course, arises as to what is a large gas engine. In 1900 a 500-h.p. engine was considered large. At the present time we have 5000-h.p. units. What is to be the large gas engine 10 years from now? The reciprocating steam engine was slow in developing. The steam turbine developed more rapidly than the reciprocating steam engine.

All are familiar with the relic in the yard of the General Electric Company at Schenectady—a 5000-h.p. unit which was one of the early turbines of this country. This was followed by an 8000-h.p. turbine and later by a 14,000-h.p. unit, and I understand that an order has recently been placed for a single unit of 30,000 or 35,000-kw. capacity. This steam turbine development has been very rapid.

If we are to install large central plants at the mines and develop electric current for long distance transmission we must have large gas engine units if producer gas is to compete with steam. A gas producer of 3000 to 4000-h.p. capacity in a single shell has recently been installed. The construction of the plant is so simple that the cost of manufacture should be low. If this producer meets the demand and proves to be a commercial proposition, there is a possibility of developing units of not less than 10,000 h.p. in a single shell (since these comments were made I understand that a unit of 75 tons

[1]This question is very fully answered by G. Neumann's article, an extract of which appeared in The Journal, Am.Soc.M.E., January 1914, Foreign Review Section, p. 09, etc.

capacity per 24 hours has been ordered), but gas engine units of a size to compete with large steam turbines are slow in developing. This central station at the mines is not pure theory. The few installations that have been made at the mines have attracted the attention of engineers and I understand that one of the large coal companies of this country is at the present time considering the installation of such a plant for the utilization of mine refuse. I was told recently that representatives of this company are in Europe seeking information relating to large by-product gas plants. I have also been told that the producer-gas interests are also alive to the situation and have sent representatives abroad within the past few weeks to study the question of large units. These large units seem essential if we are to utilize our fuel resources to the best advantage. I trust that Professor Langer can give us some definite information regarding the future of the large gas engine unit.

F. S. Giller. With reference to the point brought up about having gas-driven electric plants at the coal mines, I would like to ask whether gas engine sets, in their largest sizes, could compete with steam turbine sets, in their largest sizes, in the matter of cost of production and convenience of operation.

It is not easy to understand just what is meant by the author's statement that the gas engine is vastly superior as a fuel saver from the economic point of view. A concern handling a large power production undertaking does not consider either cost of fuel or first cost, except in their relations to the total cost of production. Undoubtedly the cost of fuel is a very important item in this total cost, but I believe that practically all of the other items, such as depreciation, repairs, rents, taxes, insurance, interest on investments, etc., are higher for gas plants than for steam plants, and that together they usually constitute a greater argument against the gas engine sets than the high heat efficiency does for them. I have lately visited many large steam-driven and water-driven electric generating plants, and also the large gas-driven plant at Gary. At this last place, I could not help being struck by the great size of the generating sets in comparison with the sizes of corresponding turbine sets, the enormous foundations and buildings required by them, and the obvious need of a large staff of men to keep them in proper repair and operation. Undoubtedly the sets would be smaller if they were using richer gas, but I doubt whether, at the present time, the best gas engine set, working under its best conditions, can compete with the best steam

turbine set, working under its best conditions. With regard to the future, the chance would seem to favor the turbine more and more, for the efficiency of the turbine is improving rapidly and continuously, whereas that of the gas engine is not so marked. The author speaks of the competition between the gas engine and the steam turbine, which lately has become very keen in Germany as well as in the United States. I am under the impression that the competition is becoming less keen and that the arguments are more and more in favor of the turbine. Several years ago, the Mond people in England were producing and distributing power gas on a large scale, but I believe their installations do not increase much, and that the progress made by gas plants generally during the last few years has not been anything like so marked as that made by the turbine, except that gas plants have been utilized in steel mills and other places where gas is to be had under unusually favorable conditions.

A comparison of the theoretical costs of production of electric energy by a large gas-driven plant and by a steam-driven plant of the same total capacity, assuming both of them to be situated at the coal pit and burning all the coal brought up from it, would be very interesting. I would like to ask whether such costs have ever been prepared and if not, whether the author considers that they would argue in favor of the gas-driven sets.

F. Z. Nedden, in reply to a question, referred to the Humphrey pump recently installed in Chingford near London. The problem of using the Humphrey pump as a generator of electricity has been taken up by the Siemens-Schuckert Works of Berlin. They are actively engaged in tests with the object of generating electricity by means of a water turbine driven by water raised by a Humphrey pump. The water after passing the turbine circulates through the pump again. So far nothing has been published as to the economy of the system. Such a plant would seem to be the proper spare for a low fall water turbine during periods of shortage in water. Instead of using a separate spare generator set driven by a steam or internal-combustion engine, the water would simply have to be pumped, by a Humphrey pump, from the tail-water back to the headwater and flow through the water turbine again, thereby saving expense for a spare generator and simplifying service.

The Author. Referring to the discussion by Mr. Freyn, while regulation by throttling the gas only will produce a smoothly running engine, I prefer to throttle air and gas simultaneously.

In reference to cylinders, cracked cylinders which have been calked have lasted a good many years and very many of them are still in use. It does not weaken or damage the cylinder if cracks are noticed soon and calked. The crack gives some relief to the lamina which is exposed to the highest temperatures and strains.

With regard to the objections in connection with the surcharging of the machine mentioned in the paper, I do not want to imply that the strains are due to the greater mechanical forces acting. There is no question that by reducing the compression pressure it is possible to work with the same maximum explosion pressure, at the same time having from 20 to 25 per cent higher mean effective pressure. The strains referred to are not caused by mechanical forces but by heating. In this matter I cannot draw on my own experience, but I have heard that the temperatures of the cylinders are increasing. This would mean an increase of strain in the cylinder walls due to the surcharging of the engines.

In reply to Mr. Nedden, I have no special information on the surface-combustion boilers. Generally it is said that the cylindrical surfaces on which the combustion takes place are filled with dust in a short time and put out of service. This, I think, is not a serious inference, since it should be possible to produce combustion cylinders at such a cheap price that it will be practicable to "scrap" them when necessary.

Professor Fernald asked about the size of gas engines. When we speak of large gas engines it means engines of 1000 h.p. and upwards. Double-acting engines of less than about 35 in. or 36 in. diameter of cylinder are not built at all, at least in Germany. The limit of the size, according to our experience there, is about 55-in. bore, by 6-ft. stroke. Tandem engines of this sort would carry a load of 3500 h.p., which means a twin unit of 6000 or 7000 h.p., about the maximum limit. This limit is not determined by shop practice or by designing, but by the impossibility of shipping by rail larger sizes of frames, cylinders and tie pieces.

I agree with Professor Fernald with regard to the production of electric power at the mines. Power can be produced at the mines for generating electricity and the current transmitted at 100,000 or 150,000 volts, which voltages I understand are being used in this country. Whether these big central stations will use gas engines or steam turbines is very hard to say; the question will probably be decided to a large extent by financial considerations.

While big units with piston engines are limited to 6000 or 7000 h.p., no limitations are given for the gas turbine, and this problem certainly represents the next advance to be made by the mechanical engineer.

In reference to waste heat boilers (in reply to a question), I cannot give the data on the necessary heating surface of these boilers. But it is not advisable to use too much of heating surface, for it is necessary to discharge waste gases with temperatures of 200 or 220 deg., in order to prevent the condensation of the steam vapor in the exhaust gases and corrosion of the tubes, as sulphur is found in any of the industrial gases.

No. 1421

EXTINGUISHING OF FIRES IN OILS AND VOLATILE LIQUIDS

Edw. A. Barrier,[1] Boston, Mass.

Non-Member

The extinguishing of fires in oils and in most of the volatile liquids has always been a difficult problem and where fires of this kind occur the results are frequently very disastrous. Our most common extinguishing agent, water, works rather unsatisfactorily upon the majority of such fires, but it is still the only one available where heroic measures are required. Comparatively recently, however, there have been two or three other materials introduced for use as extinguishers which have shown some promise for dealing with these fires, and it is the purpose of this paper to discuss these materials and the conditions under which they prove the most efficient.

2 Not all fires in volatile liquids are difficult to handle with water. When the liquid is miscible with water this extinguishing agent can be successfully used. Examples of this kind are denatured alcohol, wood alcohol, grain alcohol, acetone, etc. Where the liquid is not miscible with water little or no effect is produced except to wash the burning liquid out of the building where it may be completely consumed or, if the quantity of oil is small, possibly to extinguish the fire by the brute cooling effect of a large quantity of water sprayed upon the fire. Soda and acid extinguishers are somewhat more effective than pure water, but even they fail under most conditions. The various grenades containing salt solutions which were formerly extensively exploited are of course practically worthless.

3 The only principles that can be made use of in extinguishing fires in volatile oils are, (*a*) to form a blanket either of gas or of solid material over the burning liquid which will exclude the oxygen of the air, or, (*b*) to dilute the burning liquid with a non-inflammable extinguishing agent which is miscible with it.

[1] Inspection Department, Associated Factory Mutual Fire Insurance Companies, 31 Milk Street.

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

SAWDUST AND BICARBONATE OF SODA

4 To the blanketing type of extinguishers belongs sawdust. Paradoxical as it may seem, ordinary sawdust is an excellent extinguishing agent for certain volatile liquids, especially those of a viscous nature. A considerable number of experiments were conducted in the fall of 1912 by the inspection department of the Associated Factory Mutual Fire Insurance Companies, in the extinguishing of fires in lacquer and gasolene in tanks with sawdust, and the results were surprisingly satisfactory.

5 The liquids were placed in three tanks 30 in. long, 12 in. wide and 16 in. deep; 48 in. long, 14 in. wide and 16 in. deep; and 60 in. long, 30 in. wide and 16 in. deep. The sawdust was applied with a long-handled, light but substantially built snow shovel having a blade of considerable area. In every case the fires were extinguished readily, especially in the two smaller tanks which were about as large as any ordinarily employed for lacquer in manufacturing establishments.

6 The efficiency of the sawdust is undoubtedly due to its blanketing action in floating for a time upon the surface of the liquid and excluding the oxygen of the air. Its efficiency is greater on viscous liquids than on thin liquids, since it floats more readily on the former than on the latter. The sawdust itself is not easily ignited and when it does become ignited it burns without flame. The burning embers have not a sufficiently high temperature to reignite the liquid.

7 The character of the sawdust, whether from soft wood or hard wood, appears to be of little or no importance, and the amount of moisture contained in it is apparently not a factor, so that the drying out of sawdust when kept in manufacturing establishments for a time would not affect the efficiency.

8 It was found that the admixture of sodium bicarbonate greatly increased the efficiency of the sawdust as shown both by the shortened time and the decreased amount of material necessary to extinguish the fires. A further advantage of the addition of bicarbonate of soda is that it decreases the possible danger resulting from the presence of sawdust in manufacturing plants since it would be difficult, if not impossible, to ignite the mixture by a carelessly thrown match or any other ready source of ignition.

9 Although the efficiency of the sawdust is greatest on viscous liquids such as lacquers, heavy oils, japan, waxes, etc., in the tests referred to, fires were extinguished in gasolene contained in the

smallest tank and also when spread upon the ground. In larger tanks the sawdust or bicarbonate mixture does not work so well since the sawdust sinks before the whole surface can be covered, whereupon the exposed liquid reignites.

CARBON TETRACHLORIDE

10 In recent years carbon tetrachloride has received considerable attention as a fire-extinguishing agent. This is due largely to the activity of certain manufacturers of fire extinguishers which use liquids, the basis of which is carbon tetrachloride.

11 This substance is a water white liquid and possesses when pure a rather agreeable odor somewhat similar to chloroform. A considerable proportion of the commercial article upon the market, however, contains sulphur impurities which impart a disagreeable odor to the liquid. The substance is quite heavy, its specific gravity being 1.632 at 32 deg. fahr. It is non-inflammable, non-explosive, and is readily miscible with oils, waxes, japan, etc. When mixed with inflammable liquids it renders them non-inflammable provided a sufficient quantity is added. Its vapor is heavy, the specific gravity being about five and one-half times that of air, consequently it settles very rapidly. As an extinguishing agent it operates by both the principles mentioned in Par. 3, namely, it dilutes the inflammable liquid rendering it non-inflammable, or at least less inflammable, and it forms a blanket of gas or vapor over the burning liquid which excludes the oxygen of the air.

12 Although this paper is confined to a discussion of extinguishing fires in oils and volatile liquids, it may not be out of place to mention that the claims made by certain manufacturers producing these extinguishers which use liquids, the basis of which is carbon tetrachloride, are grossly exaggerated. These preparations, none of which is more efficient than carbon tetrachloride, are not the equivalent of the ordinary water extinguishers for general use on such materials as cotton, wood, paper, oily waste, etc.

13 On volatile liquids, oils, etc., carbon tetrachloride has, however, shown very satisfactory results under some conditions, but the readiness with which a fire can be extinguished with it depends to a considerable extent upon the skill of the operator and the nature of the fire. In tank fires the length of time that the liquid has been burning is an important factor, and in such cases where the sides of the tank become heated the only way in which the fire can be ex-

tinguished is to squirt the liquid forcibly at the sides. If the carbon tetrachloride is squirted directly into the liquid it is much more difficult, if not impossible, to extinguish the fire.

14 The height of the liquid in the tank is also a very important factor. Where the liquid is low the sides form a pocket which retains the vapor and aids considerably in smothering the blaze. When the tank is nearly full, however, this condition does not exist, and it is then very difficult, if not impossible, to extinguish a fire in a highly volatile liquid, such as gasolene; only the most skilled operators are successful in these cases. The size of the tank or the extent of the fire if upon the floor is, as would be expected, of considerable importance. In tanks larger than about 28 in. by 12 in. more than one extinguisher and operator working at a time are necessary to extinguish a fire in such materials as gasolene. In one test where a tank 60 in. by 30 in. was used no less than seven operators were necessary, and even then it was only with the greatest difficulty that the fire was put out.

15 All of the above remarks apply to carbon tetrachloride in the ordinary one-quart extinguisher as generally sold. It is probable that a large extinguisher which could throw a large stream would prove more efficient, but on account of the great weight of carbon tetrachloride such an extinguisher would have to be specially designed to make it readily portable by mounting on a truck or some similar means. Expelling the liquid by means of a hand-pumping arrangement would probably be unsatisfactory, and it would therefore be necessary to force it out in some other way.

16 A few systems have recently been installed in which an elevated tank containing carbon tetrachloride was connected with automatic sprinklers or perforated pipes located in hazardous rooms where volatile and inflammable liquids are in use. So far as is known none of these systems have as yet been called upon to extinguish a fire, but there appears to be no reason why such a system should not provide excellent protection in special cases. In such systems it would be necessary to consider the safety of the workmen and furnish ready means of escape, since carbon tetrachloride is an anesthetic and where thoroughly sprayed through the air as from an automatic sprinkler it would probably produce rapid results.

17 The nature and effect of the fumes given off when carbon tetrachloride is thrown upon a fire is a subject which has received a great deal of discussion. When the liquid comes in contact with a

fire the vapor is partly decomposed resulting in the evolution of a considerable quantity of black smoke which is undoubtedly finely divided carbon. Pungent gases are also produced which appear to be mostly hydrochloric acid with possibly a small amount of chlorine. Since carbon tetrachloride contains no hydrogen from which hydrochloric acid could be formed this substance must be produced by the action of chlorine on the gases arising from the burning material or upon the moisture of the air.

18 The fumes of carbon tetrachloride although of a very pungent nature do not produce any permanent injury under ordinary conditions where the operator can make his escape after he has inhaled all that he can stand, but they are a distinct handicap in fighting a fire and are one of the objectionable features to carbon tetrachloride as a general fire extinguishing agent. In large rooms or where a small quantity of carbon tetrachloride is sufficient to extinguish a fire the gases are of course less objectionable.

FROTHY MIXTURES

19 Another method of extinguishing fires in oils and volatile liquids which has recently been proposed and experimented with is that of using frothy mixtures. The idea seems like a very promising one and the tests which have been thus far reported indicate very satisfactory results. The idea was originated and has been developed in Germany. So far as is known no experiments have been conducted in this country.

20 The process consists essentially in causing two liquids to mix in a tank where foam is produced. The tank is made airtight and sufficiently strong to permit of the foam being forced out by carbon dioxide under pressure, and the foam is conveyed to the fire by means of a line of hose. The exact nature of the liquids has not been disclosed, but one of them probably consists of a sodium carbonate solution containing froth-forming ingredients such as glue or casein and the other an alum solution. The two on coming together generate carbon dioxide which produces froth. This froth is reported to be quite stiff and to shrink in volume but a comparatively small amount even after a period of half an hour.

21 A number of tests were conducted in the winter of 1912 in Germany; some of them on a considerable scale. In one case as much as 5 tons of crude naphtha in a tank was involved, and in another an area of 1300 sq. ft. of burning tar was used. In all cases the results

were reported satisfactory, the fires being extinguished in a short time.

22 The frothy mixture undoubtedly owes its efficiency to its blanketing action in settling upon the surface of the burning liquid, thus excluding the oxygen of the air, and to the fact that the bubbles of liquid contain carbon dioxide which upon bursting produce an atmosphere in which combustion cannot take place.

23 According to the latest reports the matter is still in an experimental stage, various details regarding the form of apparatus, most efficient pressure, and design of nozzles being under consideration; but from what has already been done it would appear that the idea is a very promising one, and that this method of extinguishing fires in oils and volatile liquids will prove to be by far the most efficient of any that has as yet been suggested.

DISCUSSION

A. E. Cluett. As I understand it, the tank was an open one, of rectangular section, which does not represent the conditions in which these liquids are ordinarily found. They are generally in a closed tank, with a bung-hole of some description, and I would like to ask whether any experiments have been made on such receptacles, and also whether experiments have been conducted where the fluid has been spread out as it would be if it were spilled over a floor.

J. Stewart Thomson (written). The author of the paper occupies a position in connection with an organization insuring sprinklered risks, which makes no recognition whatever of devices for the extinguishing of fires other than by sprinklers. His remarks are the result of some tests made by him in connection with devices other than sprinklers.

It is generally known that fires in volatile liquids which are not miscible with water cannot be extinguished by the application of water. The reason is perfectly apparent; and that the way to extinguish a fire in volatile liquids of this type is to blanket the flame or to dilute the liquid with a non-inflammable extinguishing agent, thus cooling the liquid to the point where it will not burn.

The use of sawdust in connection with fires of this character is not new. Almost every one has at one time or another used a blanket, towel or something similar for the purpose of smothering a fire. We all know that sawdust will float for a time at least on the surface of

the liquid, and that if the sawdust is put on in quantity it acts as a blanket. As a practical fire extinguishing agent, however, the use of sawdust is not practical. In a large tank of burning liquid of the type described, before the surface can be covered with the blanket, much of the sawdust will sink below the surface whereupon the exposed liquid reignites.

As to the use of carbon tetrachloride as a means of extinguishing fires in oils and volatile liquids, there are some 40 different devices on the market employing this substance. Carbon tetrachloride has been found to be remarkably efficient; a quart of this liquid contains approximately 149 cu. ft. of fire extinguishing gas. As Mr. Barrier states, carbon tetrachloride contains sulphur impurities. Its use in a metal container results after a period of time in the corrosion of the metal. Extinguishers generally employing this chemical have not been found to have long life and corrosion of the metal results in the clogging of the working parts of the extinguisher. These extinguishers have the additional drawback of operating by air pressure; it is a well-known fact that air cannot be contained over any very long period in a metal container with valves, outlets, petcocks, etc.

The manufacturers of the Pyrene extinguisher discovered the efficiency of carbon tetrachloride as an extinguishing agent many years ago, but also that this article is corrosive in its action upon metals. It was necessary to alter the nature of carbon tetrachloride in respect to this before it could be used successfully in a fire extinguisher. They also discovered that it was possible to increase the efficiency of the liquid.

While the claim is not made by the author, the inference would be from his paper that carbon tetrachloride is virtually the same as the extinguishing agent employed by the Pyrene Manufacturing Company, in his reference to "the activity of certain manufacturers of fire extinguishers which use liquids, the basis of which is carbon tetrachloride." He is probably not informed that one quart of Pyrene will generate approximately 33 $^{1}/_{3}$ per cent more fire extinguishing gases than carbon tetrachloride; therefore it will cover more fire. The one-quart pump-type Pyrene extinguisher has been approved by the Chicago Laboratories, Inc., and is in universal use throughout the country; is uniformly recognized in every state for fires of the type described by him, and that its record of efficiency in respect to these fires is remarkable.

In his statement in Par. 12, Mr. Barrier ignores the amount of

extinguishing gas referred to above. Extinguishers of this type are more efficient in cotton fires and fires in oily waste, etc., than those employing solutions of water.

In the test where a 60-in. by 30-in. tank was used, where it required a number of operators to extinguish a fire of burning gasolene, he should certainly have had no difficulty in extinguishing a fire in a tank of these dimensions if the extinguisher had been properly handled.

Mr. Barrier concludes that the use of soapsuds, described as "frothy" mixtures in extinguishing fires in oils and volatile liquids, will prove by far the most efficient of any that has yet been suggested, but statistics will not bear out the conclusion. This method has been employed to some extent on the Continent and in England for some time, and experiments have also been conducted in this country which are the subject of extensive reports. The progress of extinguishers working on this principle, however, has been inconsiderable as compared with the progress of other extinguishers, and notably, as compared with the progress made by the Pyrene extinguishers.

Henry W. Appleton. I would ask Mr. Barrier what he considers the proper proportion of soda carbonate in the sawdust, and what effect is produced on the sawdust to make it non-inflammable; that is, not only the percentage to be used to make it efficient in putting out the fire, but also the percentage to be used in rendering the sawdust itself non-inflammable.

Lewis H. Kunhardt. Mr. Thomson's statement, that the author occupies a position in connection with an organization insuring sprinklered risks, which makes no recognition whatever of devices for the extinguishing of fires other than by sprinklers, needs modification. It is absolutely at variance with the fact. Not only do the Mutual Companies give credit for many, and practically all devices which are used in extinguishing fires, but all the various sprinkler risk associations, underwriters and others throughout the country, are continuously studying all these other devices. Sprinklers are only a part of the fire protection of a piece of property. If other devices fail, the sprinklers come in and assist, and if the sprinklers should fail, there are the fire pumps, hose, etc. All the various fire protection engineers in the country are studying these problems and details thoroughly, and they do not give recognition *only* to the automatic sprinkler, although they recognize the value of it, and know it to be one of the best fire fighting agencies of today.

ALBERT BLAUVELT. As Mr. Kunhardt comments, this paper by Mr. Barrier covers a subject in itself and one which is not tied to automatic sprinkler practice. The use of hand apparatus is universal in all branches of fire protection practice in preference to waiting for sprinklers to fuse, or to waiting for hose, and the greatest success of the professional fire department so far as the majority, or some 70 to 80 per cent of the fires numerically are concerned, is accomplished with hand appliances, or usually the carbonic-acid gas extinguishers carried on the man's back as he jumps off the hose cart.

The general scope of hand apparatus is fairly broad, depending on the combustible to be dealt with and whether ceiling as well as floor fires are likely to need extinguishment. Under its title, the teaching of Mr. Barrier's paper would seem to be that this subject of handling oil or volatile or chemical fires is not a mystery. Doubtless, as suggested by a preceding speaker, the paper can be bettered by giving more detail as to the best proportions for the several useful mixtures, but the main thing is that those who will can make up effective extinguishing compounds for volatile liquid or oil fires, not perhaps of the exact alchemic virtue of the trade compounds but probably obtain twenty times the quantity at like cost.

GORHAM DANA.[1] This paper seems to cover the field remarkably well, but there is one point I would like to bring up in connection with carbon tetrachloride, and that is the effect which the gas might have on metals in the vicinity. At a fire some time ago in a garage, an attempt was made to extinguish a pail of burning gasolene with these extinguishers. They not only failed to extinguish the fire, but the gases given off were of such a corrosive nature, probably hydrochloric acid, that the damage done to the metal work on the automobiles and in the garage, was more than the direct loss from the fire itself.

F. E. CARDULLO. The claim has been made in this discussion that one quart of carbon tetrachloride at atmospheric pressure liberates a volume of 149 cu. ft. of inert gas. A hasty calculation based upon my memory of the density of air gives for the volume of carbon tetrachloride vapor liberated from one quart of liquid a value of approximately 8 cu. ft. From the data given by Peabody in his steam tables, the volume of saturated vapor liberated at atmospheric pressure from

[1] Manager, The Underwriters' Bureau of New England, 141 Milk Street, Boston, Mass.

one quart of carbon tetrachloride is 9.97 cu. ft. From data based upon the theoretical vapor density of carbon tetrachloride the volume of the vapor generated from a quart of the liquid at a temperature of 135 deg. fahr., and a pressure of one atmosphere, is 9.8 cu. ft.

It may be added that if the gas given off occupied a volume of 149 cu. ft., it would be so light as to rise in the air and would have no smothering effect upon the fire. I would suggest that, in discussion, our members ought to be exceedingly careful of their facts, and not to quote any figures, or to make any statements, of the truth of which they are not absolutely certain.

THE AUTHOR. Replying to the question of Mr. Cluett, all the experiments with these materials were tried in open containers, since that is the way lacquer is ordinarily used in most metal-working plants. The metal parts are generally dipped into the lacquer. No tests were tried on closed tanks, and it would be difficult to use sawdust and bicarbonate of soda in such a tank. It might be possible to introduce carbon tetrachloride under such conditions. One of the tests did include a fire in lacquer spread over a wooden platform which served as a floor, and the sawdust appeared to work very satisfactorily in this case.

With reference to the comments of Mr. Thomson, I would say that I recognize that sawdust as an extinguishing agent is not new, but it is not well known, and this fact is shown by the experience we have had where this material has been recommended to manufacturing establishments throughout the country. At first their attitude was one of great surprise but after they had conducted a few tests of their own, they were very well pleased with the results. This shows that even though the idea may not have been new, it was at least not at all well known and the feature of mixing bicarbonate of soda with the sawdust is a new and valuable one.

It was not my intention to mention any particular commercial device, but in view of the fact that the Pyrene extinguisher has been brought into the discussion, I will say that with reference to the matter of the corrosive action of carbon tetrachloride, tests were conducted in a Pyrene extinguisher with an ordinary garden variety of carbon tetrachloride, the usual commercial article, in which the liquid remained in the extinguisher for over a year, but as far as could be seen after using a can opener there was absolutely no indication of corrosion. In certain cases where moisture or hydrochloric acid are

present, carbon tetrachloride does cause some corrosion, but the indications are that if a reasonably pure grade of the material is obtained, there is no serious trouble from this source.

As to the comparative efficiency of Pyrene and carbon tetrachloride, a large number of tests were made to compare the two liquids when both were used under the same conditions. As far as a number of unprejudiced observers could determine, there was no difference in the efficiency of the two. It is true that Pyrene contains a few other substances besides carbon tetrachloride, among which is a small amount of gum, which by the way is combustible, and a small amount of nitro benzol which is also combustible; both of these are detrimental rather than advantageous as far as extinguishing a fire is concerned.

With reference to the proportions of bicarbonate of soda and sawdust, the ingredients are mixed in the proportion of about 8 lb. of bicarbonate of soda to a bushel of sawdust. This proportion does not render the sawdust non-combustible but it does retard its ignition. It works in the following way: If a spark or any other source of ignition comes in contact with the mixture in the vicinity of where the bicarbonate of soda is located, the latter becomes heated and begins to give off carbon dioxide. This tends to make the atmosphere in that vicinity a non-supporter of combustion and serves to extinguish or retard the fire.

In reply to the question by Professor Woolson as to whether there is any tendency toward caking after the sawdust and the bicarbonate of soda have been standing for any length of time, I may say that this feature has been pretty thoroughly tested under the worst possible conditions and after a period of six or eight months hardly any indication of caking could be detected.

No. 1422

A SYSTEM FOR THE CONTROL OF AUTOMATIC SPRINKLER VALVES

By Fred J. Miller, New York

Member of the Society

The fires in the Triangle Shirtwaist Factory and in the Binghamton Clothing Company's building, have served strongly to emphasize the fact that public sentiment is more and more placing full responsibility upon the owners of industrial establishments for the thorough protection of employees from death by fire. This responsibility is being reflected in legislation requiring the provision of suitable means of egress from buildings, fire drills, fire walls, automatic sprinklers, enforcement of rules against smoking, etc.

2 The outside fire escape, constructed in the ordinary manner, was proved in both the fires mentioned above and in numerous other fires as well to be almost, if not absolutely worthless. It is extremely difficult for a large number of employees, especially where many of them are women, to descend the ordinary fire escape, even when there is no panic and no reason for special haste. At the time of a fire it is practically useless.

3 Fire drills are probably justifiable and have in many cases saved lives. If properly maintained they are expensive, however, and in both the fires referred to above they proved worthless.

4 Some of the other things mentioned, particularly suitable fire walls, are very good, but experience seems to have proved abundantly that the best of them all is the automatic sprinkler. In theory, at least, it is supposed to be always ready for instant use, and it puts water where and when it is needed at the very beginning of a fire, before it has had time to assume threatening proportions, and usually before there is enough smoke to alarm the occupants of the building seriously or start a panic.

5 It has other important uses, such as the formation of an effective water curtain capable of arresting the progress of a fire which has

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assumed the character of a conflagration, and Mr. Albert Blauvelt and Prof. Ira H. Woolson in their papers read at the Baltimore meeting of the Society[1] present much impressive evidence of the value of sprinklers when maintained in such working order as to be available when needed.

6 The National Fire Protection Association, through its committee on safety to life, investigated the Binghamton Clothing Company's fire and in their report, commenting upon the various conditions in the building and particularly upon the entire absence of practically every efficient safeguard, say:

> Long experience and continually repeated demonstration prove that the automatic sprinkler, where properly installed and in operating condition, is always ready, operates quickly, and either extinguishes fire or holds it in check, and is the most reliable means of safeguarding lives in the majority of existing manufacturing buildings. Enclosing stairs, building fire towers, properly constructing fire escapes, etc., are all means intended to permit of escape after the fire gets under way. The automatic sprinkler almost invariably prevents the fire from assuming serious proportions.

7 This statement by an association which has made a very thorough study of the whole problem of the lessening of fire loss and the protection of lives from destruction by fire, is borne out by the experience of the Manufacturers' Mutual Fire Insurance Companies. Every risk covered by these companies must be protected by automatic sprinkler heads. It is impressive to note that the 19 companies, comprising what is known as the senior group, carry policies on industrial establishments aggregating over $2,500,000,000; employed in these factories there are, as nearly as can be ascertained, about 2,000,000 people, and so far as known not a single life has been lost in any of the buildings thus insured where the sprinkler heads were in commission at the time of the fire. A few lives have been lost in these factories, but in every such case the fire gained headway because, for one reason or another, the automatic sprinklers were out of commission at the time.

8 The general conditions respecting factory insurance are apt to surprise those who investigate them for the first time. Among the features to be discovered is that speaking broadly the stock insurance companies do not require sprinkler heads, nor more than a fractional part of the other safeguards that are required by the Factory Mutual Companies. In recent years their leading men have strongly recommended sprinklers and they give great concessions in rates upon sprinklered risks, but the local agent of an insurance company, work-

[1] Trans. Am. Soc. M. E., vol. 35, pp. 171, 231.

ing for a percentage upon the premium, virtually recommends that his own income be greatly reduced when he urges sprinklers. Their business is organized mainly with a view to simply carrying at a profitable premium rate the fire risk as it is, and save through such agencies as their Factory Insurance Association and certain committees of limited scope, they do little to lessen the fire risk and thereby the danger of loss of life.

9 In the case of a typical factory, for instance, where the local agent for a stock company receives $200 commission per year on the risk in an unprotected condition having combustible occupancy, he might receive only $10 if the same building were fully protected against fire by automatic sprinklers, etc. There is, therefore, clearly no incentive for this agent to inform the owner how his fire risk and the amount of premiums he pays for insurance may be reduced; in fact there is every commercial incentive in the opposite direction, until "mutual competition" threatens to take the business entirely out of his hands.

10 It is interesting to note that several construction companies are now offering to install automatic sprinklers without the payment of any cost whatever by the owner, the company making the installation offering to take as its pay the difference between the present insurance premiums and the lower premiums that will be paid after the installation is made, for a number of years to be determined upon. The meaning of this is simply that the owner of the building and its occupants are very much better protected against fire without any additional outlay, and that at the end of a few years the owner will have the fire protective system thus installed for his own, free of cost.

11 I do not intend to advise either for or against these companies, but cite the fact simply as conclusive evidence of the pecuniary as well as the other advantages of automatic sprinklers.

12 But sprinkler heads when once installed must then be kept ready for service when needed. Sprinkler heads are usually arranged in groups with a supply pipe for each group. In each of these main supply pipes is a valve by means of which water can be shut off from that section when necessary for repairs or by reason of the opening of one or more sprinkler heads, either in performing its normal service, or by accident. These valves cannot be locked open because it may be necessary in an emergency to close them as quickly as possible.

13 These valves introduce an element of danger and with this it is the main object of this paper to deal. The inspectors of the Mutual Companies frequently find the valves referred to closed, and the

sprinkler heads, therefore, of no more use than as though they had never been installed.

14 In an effort to lessen this evil, the Factory Mutual Companies occasionally send to their policy holders a circular letter on the subject of the danger of closed sprinkler valves. In such a report, covering the months of April, May and June 1912, 72 cases were reported where inspectors found such valves closed. In 63 of these cases the number of sprinkler heads controlled by the closed valves are given, and these foot up to 8274 heads. Assuming that the cases in which the number of heads is not given would average about the same, this means that for this report, covering three months, nearly 10,000 sprinkler heads were found out of use by reason of valves closed when they should have been open.

15 In many cases no reason was assigned for the valves being closed, which means, we may infer, that the proprietors did not know how the valves came to be closed. It is inconceivable, of course, that they would go to the expense of installing automatic sprinklers and other fire protective devices, and then allow them to be thrown out of use if they had the organization and the discipline necessary to keep such devices always available.

16 The Remington Typewriter Company owns and operates five factories on which it carries in the Mutual Companies insurance policies aggregating about $6,000,000. These factories are protected by nearly 10,000 sprinkler heads, the water supply for which is controlled by 84 valves.

17 We have had from the start a system of inspection under which a watchman at frequent intervals examines each valve to see whether or not it is open. If found closed, the reason therefor is investigated, a report is made and an effort to prevent a repetition of the error.

18 For many years no inspector of the Mutual Companies had found one of our sprinkler valves closed, until about a year ago when one was found. The reason revealed by an investigation is probably typical of a good many such cases. A mechanic had been making some repairs in a boiler house, over the boilers, and under the roof where the sprinklers were. Before going up over the boilers he closed the valve controlling these sprinklers as a precaution against accidental flooding. When he had finished the work he called down to his assistant to open the valve. He took it for granted that the assistant had done so, but for some reason the valve was not opened and the next day the inspector arrived.

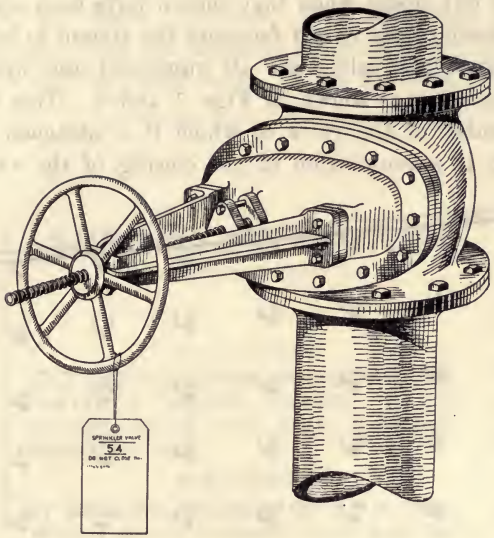

FIG. 1 SPRINKLER VALVE OPEN WITH INSTRUCTION TAG ATTACHED

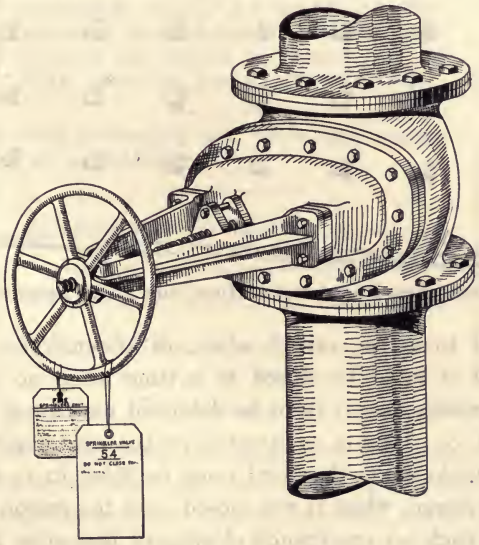

FIG. 2 USE OF RED TAG ON CLOSED SPRINKLER VALVE

19 To decrease the liability of valves being either closed without authority, or left closed when they should have been opened, we have devised and have in use in our factories the system to be described:

20 Our sprinkler valves are all numbered and upon each valve there is hung a tag as shown in Figs. 1 and 4. This tag is marked with the number of the valve to which it is attached, and, printed upon the tag is a prohibition of the closing of the valve without a

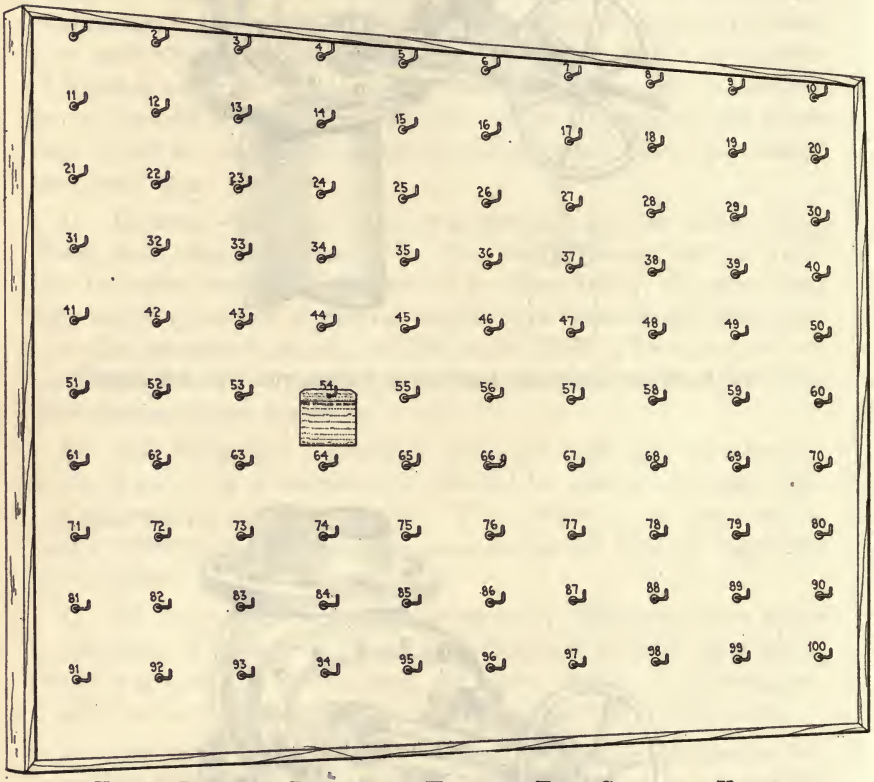

FIG. 3 INDICATOR BOARD WITH HOOK FOR EACH SPRINKLER VALVE

signed permit to do so, except when, in an emergency, or for any special reason it must be closed at a time when no permit can be obtained, in which case it is to be detached and hung upon a board, shown in Fig. 3. This is as directed by the card itself, and when it is done the blanks upon this card must be filled in to show by whom the valve was closed, when it was closed, and the reason for closing.

21 When such an emergency closing of the valve takes place, at a time when the person in charge of the board (Fig. 3) is away, the

watchman or other person having access to it goes to the board with the person who has closed the valve. Both these men then see that a red card complete but left blank is hung upon the board over the card removed from the closed valve. These are hung upon the hook

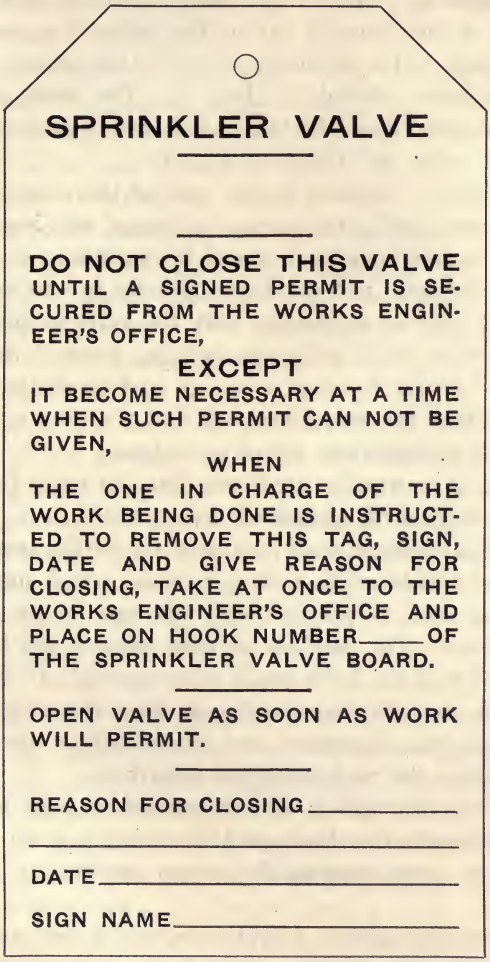
SPRINKLER VALVE
DO NOT CLOSE THIS VALVE
UNTIL A SIGNED PERMIT IS SECURED FROM THE WORKS ENGINEER'S OFFICE,
EXCEPT
IT BECOME NECESSARY AT A TIME WHEN SUCH PERMIT CAN NOT BE GIVEN,
WHEN
THE ONE IN CHARGE OF THE WORK BEING DONE IS INSTRUCTED TO REMOVE THIS TAG, SIGN, DATE AND GIVE REASON FOR CLOSING, TAKE AT ONCE TO THE WORKS ENGINEER'S OFFICE AND PLACE ON HOOK NUMBER_____OF THE SPRINKLER VALVE BOARD.
OPEN VALVE AS SOON AS WORK WILL PERMIT.
REASON FOR CLOSING_____
DATE_____
SIGN NAME_____

FIG. 4 INSTRUCTION TAG ATTACHED TO EVERY SPRINKLER VALVE

numbered to correspond with the number of the closed valve. These cards thus hung upon this hook are intended to attract the notice of the person having the board in charge, who will, after proper investigation, fill out the blanks on the red card, hang the lower portion of it on the numbered hook corresponding to the closed valve, and attach

the upper portion to the valve itself, as shown in Fig. 2, unless it shall have previously been re-opened.

22 In most cases it will be possible to obtain a written permit before closing a valve, and in such case the permit is given by the person in charge, by filling in the blanks printed upon the red tag (Fig. 5). It is then torn in two at the dotted line, and the upper portion of it given to the person applying for the permit, who attaches it to the valve before closing it (Fig. 2). The lower portion of the same red tag is hung upon the hook numbered to correspond with the number of the valve, as shown by Fig. 3.

23 The board is fastened to the wall of the room, or above the desk of the person having the system in charge, who may be the chief engineer, the superintendent, or one of his assistants, and the red tag hanging upon the hook is a constant reminder to him and a token to all others who may be interested, that the valve is closed. During this time the valve itself will have hanging upon it the red permit tag, as shown by Fig. 2, which is a sign and reminder so long as it remains there that the valve is closed and that the sprinkler heads controlled by it are therefore out of commission.

24 It will, of course, be perceived that the main feature of this system is the board with numbered hooks, which is in plain view of the factory official charged with responsibility for the sprinkler valves; a red tag upon this board is a danger signal, which means it should be taken off as soon as possible, and this can be done only by re-opening the valve. The card has written upon it the time at which it was expected that the valve could be re-opened. It is the business of the official having the board in charge, when this specified time has passed, to investigate the reason, and if a later time must be assigned to indicate it upon the card and again take it up.

25 While the description of this procedure may make it seem somewhat formidable, in actual operation it is not at all inconvenient, and it gives to some responsible person control of the sprinkler valves.

26 Of course this system of valve control is not presented under the impression that, by itself considered, it is a wonderful invention. So far as that is concerned it is no more than is being done every day for the accomplishment of various objects. It was devised to overcome a difficulty which, in view of all the conditions now existing, it seems rather important to eliminate in factories generally where sprinklers have been installed; it can save property from destruction by fire and may easily save the lives of those employed in factories; for

that reason it is here presented. It was not devised by myself alone, but, in accordance with the regular practice in our organization, the idea of it was presented to all our factories and it was then developed and perfected by the suggestions of the superintendents of those factories, Messrs. E. E. Barney, G. B. Brand, W. N. Brand, all of whom are members of the Society, and Mr. C. W. Burges.

27 Mr. John R. Freeman thought so well of it that it has been recommended by the Mutual Companies, including those of which he is president, and his associates have added some improvements to the red card, notably by arranging it so that the coupon which has been hung upon the board to signify a closed valve goes, when the valve is opened, to the superintendent, who is thus enabled to know how much of valve closing is taking place, and the reasons for it. This, I believe, is an excellent feature. The cards are now being supplied to all policy holders in the Mutual Companies and a reproduction of them is shown in Figs. 5 and 5a.

28 In some of our factories, there are certain valves which, with the approval of the mutual companies are closed during the winter months. As it was thought to be undesirable that a red card should hang upon the board, as shown by Fig. 3, for some months, because danger signals constantly displayed are apt to cease to convey the idea of danger, they have proposed in one of our factories to cover the red card hung upon the board under such circumstances by a green card. That may be the best way to do it, but I am inclined to think that a better way is to leave off the red card entirely from the board and to put a memorandum into a tickler file so that the matter of reopening the valve will be automatically presented to the person in charge upon a predetermined date. This memorandum may appropriately be one of the white cards shown in Fig. 4 properly filled out; but the red tag should in all cases be hung upon a closed valve.

29 Mr. Freeman does not think so well of the white card. We use it because by means of it, always hanging upon a valve, a person whose business it may be to close a valve can learn and become familiar with what else is required of him besides merely closing a valve and then perhaps forgetting it. In the interest of the life-saving feature, the Remington Typewriter Company will supply such cards free to all who may wish to use them. Some are printed upon cloth for use when valves are out of doors, others upon cardboard. With the cards thus supplied it will be necessary only for the factory to provide for itself, the board with ordinary screwed brass hooks, numbered as shown—to put the board up before the eyes of a responsible factory

(SEE OTHER SIDE)

FIRE

SPRINKLERS SHUT

Attach this Tag to Valve

Valve No..... { Every Sprinkler Valve should be numbered, preferably with white paint figures at least 2 inches tall. May be on pipe close to valve.

Shut on.....191 .....M.
Day of Week Month Date Hour

By.....
Name of man who shut the valve

To be shut Only until.....191 .....M.
Day Month Date Hour

Reason for Closing.....

.....

Closing Authorized by

Tag No..... .....
Signature of Master Mechanic or other official

Don't remove this tag until valve is opened. If valve is not opened at above time, fill out and attach a second tag extending time; also a third tag if need be.

Opened On.....191 .....M.
Day Month Date Hour

By..... { After opening valve take this tag to Master Mechanic's Office to be kept until next Insurance Inspector's visit.
Signature of man who opened valve

This Coupon is to be torn off and hung on conspicuous peg board in Master Mechanic's Office until tag is returned signed by the man who opened valve. After return of tag, enter date of opening on this coupon, and send it to General Superintendent's Office.

FIRE SPRINKLERS ARE SHUT OFF

In.....
Name of Building and room

At Valve No.....on.....191 .
Day Month Date

By .....
Name of man who shut the valve

To be shut Only until.....191 .....M.
Day Month Date

Reason for Closing.....

.....

Closing Authorized by

Tag No..... .....
Signature of Master Mechanic or other official

Opened by.....on.....191 .
Name of man who opened valve Date

(SEE OTHER SIDE)

FIG. 5 FORM OF RED CARD USED BY THE MUTUAL COMPANIES

(SEE OTHER SIDE)

The danger of greatest loss from Fire in factories insured in "The Mutuals" to-day comes from Sprinkler Valves improperly closed or forgotten.

On almost every day the insurance inspectors find somewhere an important Sprinkler Valve shut. Often this occurs in factories which take pride in their order and discipline.

RULES.

1. Sprinkler valves must not be closed without authority from the Master Mechanic or other official, except in case of accident, when notice must be sent at once to the Master Mechanic.
2. A tag must be filled out and attached to the valve (inside or yard) every time a valve is closed.
3. Never shut off more sprinklers than necessary, and keep off only for shortest possible time.
4. Provide for immediate turning on of water in case of fire. Where large values are involved keep a man near the closed valve ready to open it on notice.
5. For long and important shut offs provide extra watchman to patrol shut off area and notify Insurance Companies.

The Master Mechanic or Superintendent should provide in his office a conspicuous board with hooks, for these danger signals, and with supply of extra cards. Preferably a white board with a red border marked at the top—

"NOTICE OF FIRE SPRINKLERS SHUT OFF."

This method of Coupon Danger Cards was invented and tested in the Remington Typewriter Works and appears to be the best yet devised. Its adoption everywhere is urged by the Associated Factory Mutual Fire Insurance Companies.

A supply of these tags should be kept in Master Mechanic's Office, and may be obtained free of cost from the Associated Factory Mutual Fire Insurance Companies, Inspection Department, 31 Milk Street, Boston.

A few spare tags for emergency use should be kept conspicuously placed near desk of head of each department.

FIG. 5a REVERSE OF MUTUAL COMPANIES' RED CARD

official, and then to see that the system is used. And it is safe to say that if an owner finds that this or some such system cannot be maintained in effective operation, it may be taken as a symptom of the need of better organization in that factory.

DISCUSSION

George I. Rockwood. As Mr. Miller points out in his very interesting paper, the possibility of the unauthorized closure of automatic sprinkler system shut-off valves is the only remaining nightmare before the eyes of fire protection experts. In view of the record of fire losses due to such unauthorized or accidental shutting off of the water supply to sprinkler systems, it is singular that so little has been done to give any *automatic* protection against such a possibility. All the rest of the apparatus, such as the sprinkler head, the alarm valve, the dry pipe valve, and the system of sprinkler piping, has been developed to a high pitch of perfection; but this one point remains to be developed.

Impressed with the importance of this problem, I have invented and developed a simple and inexpensive piece of mechanical and electrical apparatus designed to supplement the very suggestions contained in Mr. Miller's paper. Mr. Miller desires the master mechanic, or the person in charge of the sprinkler systems in an establishment, to divide a printed card in the middle, fill out on each half the reason for closing the shut-off valve and the time when it may be expected to be opened and the pressure restored to the system again, and then requires him to tie one-half of the broken card to the stem of the gate valve and to carry the other half into the office and leave it with some authorized person who shall share with him the responsibility for shutting off the water supply.

This system would be, of course, entirely satisfactory provided the person who is required to carry it out really does so. It does not take very much imagination, however, to suggest many everyday reasons why it would be for the interests of such a person to omit such notice altogether, in which event there is a large possibility that the valve may be left shut, whether intentionally or inadvertently, for a long period of time. For example, a workman may go home to luncheon at noon leaving the valve shut, and something may interfere with his return; if the job is to be done at night, or the job of repairing the sprinkler system extends into the evening, the person notified by the other half of the card may be absent until the next day, in which case

it might easily be that the valve would be left shut at least until his return.

The device which I have developed makes it *necessary* for the master mechanic to attach his card directly to an electric switch that controls an electric bell, which switch must be opened before he can attach it, and cannot be left open without the aid of a second person. This second person should be the one relied upon to see that the valve is actually reopened at the appointed time. He is also the one who would pay attention in case the valve was shut by some unauthorized person who had not first gone through the process of notifying the man in the office, and getting his assistance in holding the switch open, as this would result in turning in an alarm directly in the office. The way this is effected is by attaching an electric circuit closer to the valve stem of the main shut-off valve in a way such that any motion of the valve stem in the direction of shutting the valve will immediately cause the circuit closer to close the electric circuit through the alarm bell and the red light in the office.

When the switch is opened through an arc of 180 deg. it will be held open provided the master mechanic has caused the circuit closer on the valve stem to close by shutting the valve; but this would result in an alarm in the office if the assistance of the man who is to be notified that the valve is to be shut has not actually been obtained to hold the controlling switch clear open against an electric magnet energized by the same current that lights the light and rings the bell, while the other man goes to the sprinkler riser and shuts the valve.

It will be observed that this mechanical and electric supervisory system is entirely supplementary to the system proposed by Mr. Miller, that it does not interfere with it in the slightest degree, and that its influence is positive if it actually works as intended, but that if it should fail to work there would be no interference with the operation of the simple card scheme. This system has been in operation in my own factory for several months, has been demonstrated to various underwriters, and seems to be quite satisfactory. No doubt about its operation, either theoretically or actually, has thus far developed.

The principle covering the development of this device is that more than one man should have something to say about the opening and shutting of sprinkler valves, and that the operations of the extra person should be mechanically interlocked with those of the person directly charged with the oversight of the sprinkler system. The device in question, in addition to being supervisory over the motions

of the shut-off valves, is also a fire alarm and a water pressure alarm. In regard to the latter point, it may be said that the bell will ring and the light will light if the water pressure falls beyond a certain predetermined point.

In the history of sprinkler systems it has frequently happened that main shut-off valves have been shut with a heavy hand and then *apparently* opened again, when, in reality, it was subsequently found out that the only part of the valve which opened was the stem, which had become detached from the plug. This may easily happen when the valve stem is shut with a big wrench in an outside gate and the stem is twisted after the valve has bedded upon its seat. In such a case as that, my alarm system would call the fact to the attention of the office automatically.

James P. Tolman. It appears to me that Mr. Miller's instruction tag does not afford all the protection that is possible. In our mill we have used a tag similar in style to that, but in place of the circular hole we have a slotted hole, through which is passed a piece of lace leather. The leather is also passed through the yoke and the arm of the hand wheel, which seals the valve. The lace leather is secured by a tubular rivet, easily applied. Each valve which controls sprinklers throughout the mill is always sealed, either open or shut. If it is a valve which should be closed in winter, it is sealed shut. If it is a valve which should be open at all times, it is always sealed open, and the tag requires any person breaking the seal to take the tag to the office. Recently the Mutual Insurance Companies have been sending out the tag shown in the latter part of the paper, and we felt that it was an improvement and adopted it. But we also retain our old form of tag, which we now make from green card, and this green tag is on every normally conditioned valve at all times, showing that the valve is sealed, and that persons who break the seal of the valve are to take the tag immediately to the office. If a valve has its seal broken, there is a notice. The red tag has supplanted the green tag, and every valve should always have one or the other tag upon it.

One of the objections made by Mr. Rockwood to the tag is that when a valve was marked with a red tag, it would never be seen by anybody unless he *happened* to see it. I want to say that it is the practice of all the mutually insured mills to have weekly inspections of the fire protective system. We have different men inspect the system for different periods of time. If a valve is wrongly marked,

having the red tag instead of a green tag, it will be brought to the attention of the superintendent at least as soon as Monday morning.

GORHAM DANA.[1] I agree with the previous speakers that the matter of a closed valve in the sprinkler system is a most vital point in the matter of fire protection in sprinkler-equipped plants. If we could eliminate the unnecessarily closed valve, the fire loss would be very materially reduced.

The scheme which one of the speakers proposed is an admirable one for certain conditions, but it does not seem to cover our conditions. In order to succeed with such a plan, it is necessary that a plant should be extremely well managed, with an efficient master mechanic and an efficient superintendent. There are a good many plants equipped with automatic sprinklers that do not have these conditions, a great many tenant factories, and a great many New York loft buildings, where there are many different firms in the building, and where it is difficult to get at any one to care for the sprinkler system. There are a great many small plants in various towns and cities that have no master mechanic who is worthy of the name, where the problem is very different.

The organization with which I am connected has been working on this problem for a number of years, and we have solved it in rather a different manner, which, to my mind, covers cases which this scheme would not cover. The plan is to have the valve fastened by the insurance inspectors. The tag which is used (Fig. 6) is slipped over the seal and is used in case the seal is broken to notify the insurance company. That is to say, the tag is similar to the one shown here, but somewhat differently worded, and in case the seal has to be broken to close the valve, the person who breaks the seal fills out certain data as to the circumstances (the reason for closing the valve, etc., the present condition of the valve, whether it was opened at once or not) and this information is put on the back of the tag and the tag mailed to the insurance inspection department. We find it to be extremely necessary in any plant which has not an efficient management, to have this information sent to the insurance interests.

Mr. Rockwood's scheme is a very excellent one, and I trust it will come into general use, but it has, however, one drawback: if a valve is closed for any length of time the alarm does not do much good. If

[1] Manager, The Underwriters' Bureau of New England, 141 Milk Street, Boston, Mass.

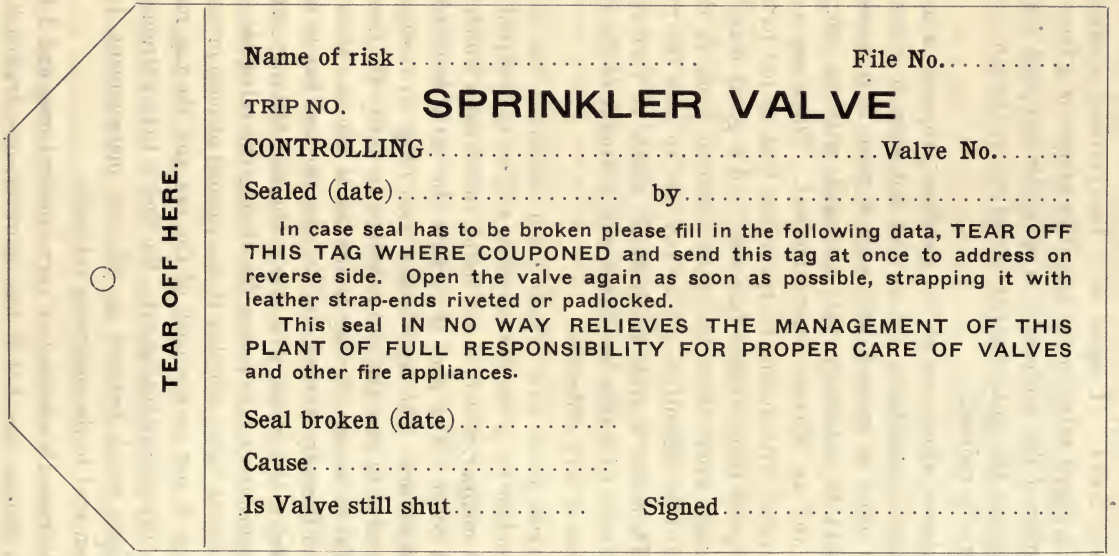
TEAR OFF HERE.
Name of risk..... File No.....
TRIP NO. SPRINKLER VALVE
CONTROLLING.....Valve No.....
Sealed (date)..... by.....
In case seal has to be broken please fill in the following data, TEAR OFF THIS TAG WHERE COUPONED and send this tag at once to address on reverse side. Open the valve again as soon as possible, strapping it with leather strap-ends riveted or padlocked.
This seal IN NO WAY RELIEVES THE MANAGEMENT OF THIS PLANT OF FULL RESPONSIBILITY FOR PROPER CARE OF VALVES and other fire appliances.
Seal broken (date).....
Cause.....
Is Valve still shut..... Signed.....

FIG. 6 FORM OF REPORT TAG IN USE BY UNDERWRITERS' BUREAU OF NEW ENGLAND

a valve is closed for repairs, for an hour or two hours, you have got to shut off the alarm and cannot have it ringing all the time, and for that reason some notification to the people most vitally interested, the insurance companies, is most desirable in my opinion.

W. H. Kenerson (written). There can be no difference of opinion in regard to the necessity for keeping all sprinkler valves open, and if it is impossible to provide some of the mechanical supervisory devices, the scheme outlined in the paper is highly desirable. I take it that the purpose of the paper it to put before the public a scheme which anybody can employ, and the description of the system is in such detail that I beg to suggest one further simple expedient which will make it more effective. The majority of us have probably had difficulty, when using a system employing tags hanging on open hooks, in keeping the tags in place because of drafts, careless handling, etc. The simple expedient of turning the hooks upside down, will almost entirely obviate the difficulty. Such an arrangement would make this system much more reliable.

Charles H. Bigelow. While I believe the tag is a good thing, exceptional discipline is required to keep it up in every case, especially in places having only a small mechanical force of the class often employed, and where the office is a long distance from the valves. There the tendency would be, especially for a short job, for the man to shut off the valve and do the work, instead of taking the tag to the office, returning to do the work and then going back to the office after the tag, particularly as many of them regard such regulations as "red tape" if they think of them at all. The trouble is, however, that while the men may open up the valve in most cases, there is always the chance of the man being called away and forgetting to open the valve, so that probably the tag system, even if not kept up the way it should be, would be a help.

Pomeroy W. Power (written). The weekly inspection seems to be a very good system to use in connection with any method of keeping track of sprinkler valves. Our plan is giving good results. Every Thursday an inspector examines each indicator post gate and signs a tag, which is attached to the post, with his name and the date. After his examination of all sprinkler valves, the inspector signs a typewritten form of letter, which reads something like this: "I have this day inspected all of the sprinkler valves. They are all open and

TABLE 1 INSPECTION RECORD OF HOSE HOUSES AND SPRINKLER VALVES

Andrew McLean Co., Passaic, N. J.

	Location	Sizes In.	State of Valve
1	Third St., gate valve, city connection	6	Open
2	Mill No. 1, napping depart., indicator post	6	Open
3	Mill No. 1, napping department, drip	2	Closed
4	Hose house No. 1, two-way hydrant, ft. hose spanners, washers, nozzle.		Clear
5	Boiler house, sprinkler line	3	Open
6	Boiler house, sprinkler drip	1½	Closed
7	Boiler house, 6-in. gate in line to mill No. 1	6	Open
8	Old boiler house, drip	2½	Closed
9	Old boiler house, sprinklers under gallery	2	Open
10	Kier room, kier dye and breach room sprinklers	3½	Open
11	Kier room, drip for above	1¼	Closed
12	Old grey room, drip	2	Closed
13	Mill No. 1, second floor sprinklers for tower	1¼	Open
14	Mill No. 1, indicator post (tower entrance)	6	Open
15	Mill No. 1, indicator post (shipping room)	6	Open
16	Drip in shipping room	1¼	Closed
17	Hose house No. 2, one-way hydrant, ft. hose spanners, washers, nozzle.		Clear
18	Storehouse No. 2, indicator post	6	Open
19	Storehouse No. 2, basement drip	2	Closed
20	Office, indicator post	6	Open
21	Office, drip in pit of 1st toilet	2	Closed
22	6-in. valve in main rear shop	6	Open
23	Storehouse No. 3, basement drip	2	Closed
24	Storehouse No. 3, two-story end, indicator post	6	Open
25	Storehouse No. 3, two-story end, drip	2	Closed
26	Storehouse No. 3, one-story end, indicator post	6	Open
27	Hose house No. 3, two-way hydrant, ft. hose spanners, washers, nozzle.		Clear
28	Hose house No. 4, two-way hydrant, ft. hose spanners, washers, nozzle.		Clear
29	Hose house No. 5, two-way hydrant, ft. hose spanners, washers, nozzle.		Clear
30	Mill No. 2, indicator post	6	Open
31	Mill No. 2, drip	2½	Closed
32	Hose house No. 6, two-way hydrant, ft. hose spanners, washers, nozzle.		Clear

The above has been examined by me to-day.

Date,..... Signed,.....

Countersigned by.....

in good order, except——" Under the list of exemptions he fills in the location of any valve that is closed, the date when it was closed and the date that it is expected to be opened. That letter, after being signed by the inspector, is sent to the manager as a weekly report. After about four years use of this plan, we have had only one case of a valve being found closed.

LEWIS H. KUNHARDT. This whole question comes down to a matter of responsibility. Some one must be made absolutely responsible in a plant for the maintenance, for instance, of the sprinkler valves, the same as some one must be made responsible for the maintenance of any other piece of apparatus in the plant. A combination of the inspection and the tag will, we believe, fulfill these conditions of keeping the valves open. Of course, there is a fair possibility that something may slip up, but when it does slip up, the point is that you must have some responsible person to whom to look to see why it occurs, and that it does not occur again. Any one who tries this combined plan, that is, a system of inspection and tagging the valves, will accomplish the result.

F. J. BRYANT (written). At our plant we have inspection cards giving in itemized detail the route to be followed, showing the location, size, type and duty of each valve, the location of each length of hose, each nozzle, tool, etc., as shown in Table 1. The man making the inspection takes the card and checks each item as he comes to it (if he finds it correct) and notes any discrepancies on the margin. After completing the round he signs the card, as the previous speaker said, and places it upon the manager's desk, who countersigns it and has it filed for further reference.

THE AUTHOR. Before adopting this system of cards, we carefully considered mechanical and electrical devices for controlling our valves. There are men in our organization who have devoted a great deal of study to electrical devices and they at once thought of like means of controlling valve opening and closing, and proposed several schemes. We have, after careful study of the matter, come to the conclusion that it is better to depend upon our discipline and organization and to have some one responsible for these valves. If we put a mechanical or electrical device on the valves, any one would still be able to close it at will, and it would not involve the doing of anything else. We believed that notwithstanding the use of such a device we would have

to go to some form of tag and report in addition. It is a fact that any automatic device, such as a water level indicator in a boiler, or anything of that sort, that is depended upon to indicate danger or abnormal conditions is apt to be depended upon too much; some day it will not work and then there is disaster. For our conditions at any rate, a thoroughly organized system for taking care of these valves and a definite placing of responsibility upon someone for their condition is the best. The seal idea for the card seems to be good, and would be a useful addition to it.

We had, as stated in the paper, before this card system was installed, a very thorough system of inspection for these valves, and we still maintain that inspection system. But the trouble about a weekly inspection is that if the inspector looks at a valve on a Thursday morning, the valve may be closed until the next Thursday morning and no one know anything about it except the man who closed it. Our system provides that someone must know about it; some responsible person must know immediately if the valve is closed, and we think that is important. Of course, there must be discipline, a definite placing of responsibility and strict accountability.

No. 1423

THE NEED OF MORE CARE IN THE DESIGN AND CONSTRUCTION OF ELEVATED TANKS

By W. O. Teague, Boston, Mass.

Member of the Society

The elevated or gravity tank for fire protection systems has been from the first an important limited secondary source of water supply, and its value has increased greatly with the increase in number and size of tanks installed generally throughout the country, especially in those cities and districts where the public water supply is of low pressure, as is the case in Philadelphia. The tanks are usually located above buildings in cities and on detached towers in the country. A typical detached structure is shown in Fig. 1, which is a 150,000-gal. tank on a 242-ft. tower at the plant of the New York Shipbuilding Company, Camden, N. J.

2 The tanks were first made of wood, but there are now as many being made of steel. Wooden tanks have been built up to 100,000 gal. capacity, although they are rarely larger than 60,000 gal., for above this capacity the steel tank is cheaper and more practicable. The cost of a 60,000-gal. tank of wood or steel erected on a 75-ft. steel tower is about $3,000. Steel tanks are built in large sizes, one of the largest being of 1,200,000 gal. capacity; this one is 50 ft. in diameter and 90 ft. high, and is supported by a steel tower 130 ft. high.

3 Failures of tanks in service, involving loss of life and destruction of property, have shown the need of more care in the designing and construction of them. To insure the best results, the following features should have attention.

WOODEN TANKS

4 The tightness and durability in the wooden tank depends chiefly upon the quality of the lumber and the details of its construction. Selected tank stock only should be used consisting of white cedar, cypress, white pine, Douglas or Washington fir, or redwood, and the lumber should be free from sap, loose or unsound knots, worm-holes

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and shakes, and be thoroughly air-dried. Both the staves and bottom are usually made up of 2½-in. stock dressed both sides, for tanks up to 16 ft. in diameter and 16 ft. deep; for larger tanks 3-in. stock is used. Planks for this purpose should be full length without splices.

FIG. 1 TYPICAL DETACHED TOWER STRUCTURE FOR ELEVATED TANK. CAPACITY 150,000 GAL.; ELEVATION 242 FT. TO BOTTOM OF TANK

5 The strength of the wooden tank depends principally upon the size and spacing of the iron hoops. The importance of the matter of the hooping will be appreciated when it is realized that the overstressing of even one hoop may result in the bursting of the tank. The wooden tank being originally merely a development of the barrel where flat hoops were necessary to permit of tightening by driving them toward the enlarged middle, it was natural to use also flat hoops for the tank and the tank was also made tapered so that the hoops could be tightened by driving, although later they were tightened principally by hoop lugs. It was claimed that the tapered shape had also the advan-

tage of preventing the hoops from dropping down over the tank, if it was allowed to remain empty and the staves to shrink from drying.

6 The tapered shape of tank is not important, however, since a tank which has been allowed to dry up, has been seriously damaged thereby and cannot be made tight without extensive repairs, sometimes necessitating the rebuilding of it. In fact most tanks are now made without taper and the hoops are found to remain where placed. The tapered tank costs somewhat more to build since the staves must be fitted more carefully and the design undoubtedly would have been entirely discarded long ago, except that some architects and purchasers believe a tapered tank presents a more pleasing appearance. The amount of taper is so small, being usually 1 in. per ft., thus giving a batter of ½ in. per ft. to each side of tank, that its absence is hardly noticeable except on very high and small diameter tanks. The only objection to the tapered tank, however, is its extra cost.

FIG. 2 APPEARANCE OF INSIDE SURFACE OF FLAT BAND HOOP THAT RUSTED THROUGH AND FAILED ON WOODEN TANK

7 In the early studies of this subject many serious failures of tanks were traced to weakening of the flat hoops by their rusting at the back where they bore against the staves, due to moisture from rain being retained between the surfaces of the hoop and staves. Fig. 2 shows the appearance of the inside face of a flat hoop which rusted through and allowed a tank to burst. These failures were largely unpreventable, as it was difficult to inspect properly the condition of the hoops, and also impossible to paint them while the tank was in service. The use of hoops of round rod without welds has remedied this trouble as their surface is nearly all exposed for inspection and painting, and also they are not so subject to corrosion since the exposed surface of a round rod is much less than that of a flat bar or band of the same cross sectional area.

8 Round rod hoops are now being used exclusively on well built sprinkler tanks, and the flat hoops of a large percentage of previously built tanks have been replaced with them. It is, of course, advisable

that the remaining flat hoops also be replaced, but if this is not done, the hoops should be carefully examined every few years to note their condition. In making such an examination, a round rod hoop should be placed on each side of one of the lower flat ones, within easy reach from the balcony; the flat hoop should then be struck smartly with a pointed hammer at intervals of a few inches all around to detect thin places, or else the hoop should be removed and examined. Examination at wider intervals is not sufficient since the hoop may be in good condition at one point and be nearly rusted off at a point six inches away. If this lower flat hoop is found to be corroded materially, all the hoops should be replaced with round rod ones, or, if preferred, the flat ones may be retained and round rod ones placed between them.

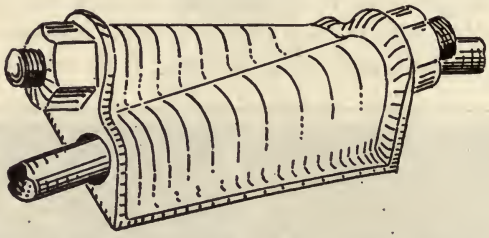

FIG. 3 PREFERABLE DESIGN FOR END LUGS IN MALLEABLE IRON FOR ROUND ROD HOOPS

9 Another point of weakness in the flat hoop is at its connection to the cast-iron lugs which is usually made by riveting. The use of round rod hoops, however, permits of a satisfactory connection to the lugs, but at first many tank failures resulted from the use of light cast iron lugs. These are now made of malleable iron, the best design being shown in Fig. 3. The hoops are so placed on the tank that the lugs do not come in a vertical line.

10 Round rod hoops are so spaced that the stress will not exceed 12,500 lb. per sq. in. when computed from area at the root of the thread. The proper spacing can readily be found from the following formula:

$$\text{Spacing of hoops (in.)} = \frac{\text{Safe load for given hoops (lb.)}}{2.6 \times \text{diameter (ft.)} \times \text{depth (ft.)}}$$

The depth used is the distance from overflow to point where hoop is to be located. The top hoop is placed 2 in. from the top of the staves and the spacing between hoops should in no case exceed 21 in. An extra hoop or two is placed at the croze to take the additional strain due to the swelling of the bottom planks.

11 The tank roof, since it in no way serves to retain the water, has usually been nothing more than a make-shift cover. In the early days a single flat roof was used on outdoor tanks, but this held the snow and ice and required strong joist supports to keep it tightly in place. The snow also interfered seriously with the opening of the hatch giving access to the interior of the tank. A conical roof was then built over the flat one which remedied these difficulties. It also greatly increases the efficiency of the roof in preventing radiation of heat from the tank water in winter as it provides a dead air space between it and the flat roof in addition to the one between the latter and the water, thus reducing the cost of heating in freezing weather. The conical roof also gives a better appearance to the tank top. A

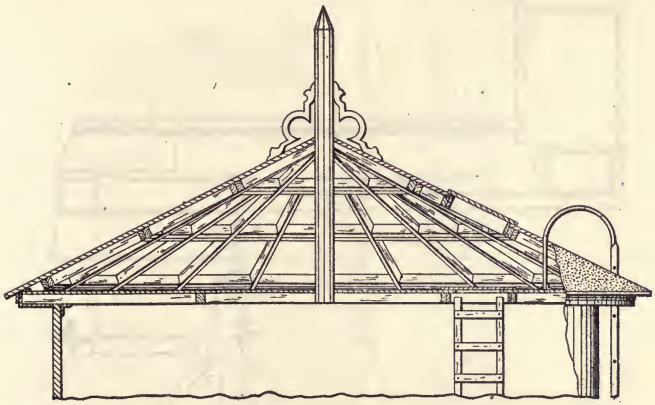

FIG. 4 A GOOD DESIGN OF DOUBLE ROOF FOR ELEVATED TANK

well built roof is shown in Fig. 4. It should be tightly fitted around the tank top to maintain the dead air spaces.

12 The roof covering which was most generally used was the wooden shingle, but these were found to catch fire from burning embers which spread to the tank staves, weakening them so that the tank burst. Non-combustible coverings, such as galvanized sheet iron and composition roofings are now used.

13 Much trouble has resulted from leakage in the wooden tank, because it has not been firmly supported. The wooden tank is locally weak, not being of unit construction, and the lack of firm support has permitted working of the joints. It is supported only from the bottom, none of the weight being carried by the staves. Wooden beams were first used as supporting members, and these were placed on the

roof of a building or tower as a grillage, and the tank bottom set on them. In time the wood rotted because of moisture from the tank bottom, permitting the tank to settle and causing leakage; there was also danger of collapse of the tank because of this weakening of the joints. The use of steel I-beams as grillage members as shown in Fig. 5 avoids these difficulties. The beams should not be spaced over 18 inches clear between edges of flanges, and the tank bottom is placed directly on the steel.

STEEL TANKS

14 When good tank lumber began to get scarce and to increase

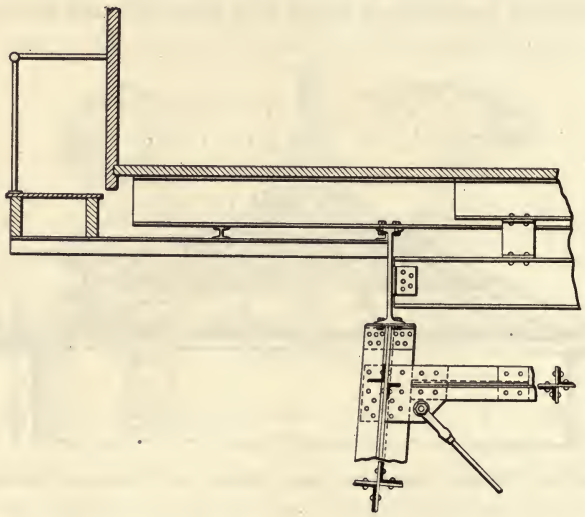

FIG. 5 DETAIL VIEW OF ARRANGEMENT OF STEEL I-BEAM GRILLAGE FOR SUPPORT OF TANK

considerably in price, boiler and iron companies turned their attention to the manufacture of steel tanks and towers, especially of the larger sizes. In the design and construction of these tanks, the manufacturers have drawn on their extended and complete experience in boiler and bridge work, but the matter of supporting the tank properly had to be experimented with until experience with tanks in service was available. The simplest form of steel tank is the flat bottomed one and tanks of this type give satisfactory service, provided the bottom is supported by a steel grillage as in the case of the wooden tank. One possible source of trouble is from corrosion of the bottom, and to prevent this in so far as possible the bottom plates are made

somewhat thicker than is necessary for strength alone, and the grillage I-beams are of a height and spacing to permit of inspection and painting of the bottom. When the tank is to be placed on a concrete tower, it may rest directly on a reinforced-concrete slab with the bottom thoroughly grouted in place with neat cement.

15 The preferred form of a tank to be placed on a steel tower is that having a hemispherical or elliptical bottom. The construction in this form is cheaper than for the flat bottomed tank as the bottom is self supporting and a steel grillage is unnecessary. The entire bottom is also accessible for inspection and painting, and corrosion is reduced to a minimum since the plates are exposed to the air.

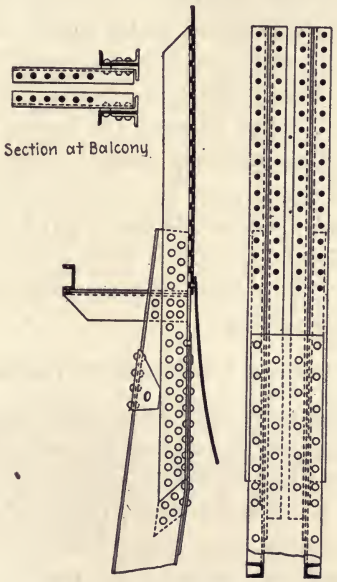

FIG. 6 DETAIL OF ATTACHMENT OF TANK SHELL TO TOWER POST AND CIRCULAR GIRDER CONSTRUCTION

16 Plates for use in steel tanks are made somewhat thicker than is necessary for strength in order to make them durable against corrosion. The minimum thickness is ¼ in., except that ⅛-in. plates are used for roofs. The plates composing the lowest cylindrical ring are 5/16 in. thick for 60,000-gal. tanks and larger, and the bottom plates 5/16 in. thick for tanks 75,000 gal. and larger.

17 One of the weaknesses of steel tank construction in the past has been poorly designed connections of the tank shell to the posts of the supporting tower. When the posts have a batter, as is usually the

case, the inward thrust due to the horizontal component of the weight is provided for by a circular girder consisting of ¼-in. plate 24 in. wide, attached to the lowest cylindrical ring by an angle and stiffened by angles or a channel at the outside edge, as shown in Fig. 6. The posts also connect to the tank shell at this point and the design is such that the load will be transferred from the shell to the center line of the posts so as to avoid eccentric loading. A number of tanks built without circular girders have failed by the posts crushing in the tank plates. Others with the girder, but having eccentrically loaded connections to the posts, have failed by bending of the upper posts.

18 As the hydrostatic pressure on the tanks is comparatively small, it is not necessary to provide standard riveting for the thickness of plates used. The joints of the plates should be riveted so that the unit stresses on the net section of the plates and rivets will not exceed 7500 lb. for shearing and 20,000 lb. for bearing. The horizontal joints are single lap riveted, except between the lowest cylindrical ring and the bottom, which are double lap riveted. The vertical joints also are single lap riveted except those in the lowest cylindrical ring, which are double lap riveted. The rivets are entered from the outside and driven from the inside and the inside seams calked. One of the strong features of the steel tank is that when once made tight, it gives practically no trouble from leakage.

TOWERS FOR ELEVATED TANKS

19 Towers to support wooden tanks were originally built of wood. While the tanks were small in size, say up to 20,000 gal. capacity, and elevated to a comparatively moderate height, this construction proved fairly satisfactory. With the increases in size of plant buildings and extensions of them, considerably larger tanks and higher towers were required, and the builders, realizing the inadequacy of wooden construction under these conditions, began to make towers of steel. One manufacturer built towers of iron pipe construction having from four to twelve posts and the members connected by cast iron fittings, but it was practically impossible to provide resistance against uplift due to wind pressure, and with the tank empty the structure was in great danger of overturning. The inside of the pipe was subject to corrosion and as its condition could not be inspected or the surface painted, the metal might be greatly weakened without its showing on the outside. These towers were sold largely in a knocked-down condition by mail order and erected by the purchaser who of course could not be expected to have expert knowledge as to their construction. Ob-

viously, these towers should not be depended on for such an important use, and it is advisable to replace those already built with well-designed structural steel towers to forestall failure.

20 The wooden tank manufacturers' shops were originally fitted only with woodworking machinery, so that it became necessary for them to prepare for the fabrication of steel work. The managements not being experienced in structural steel designing, naturally selected the simplest design possible for the towers The posts and girts consisted usually of two angle irons, placed apex to apex and strapped together at intervals of several feet by tie-plates shop riveted to the angles. The column sections were spliced by angles which were shop-riveted at one end to the post; the other end was field-bolted in erecting the tower, as this was the simplest form of connection and the easiest one to make. Furthermore, it had the advantage that the bolting could be done by the regular erecters which made it unnecessary to have first-class mechanics in the erecting gangs and to carry special tools. This, however, was not good construction and the manufacturers are now field-riveting these connections.

21 The struts were at first connected directly to the posts by bolts. This construction is objectionable because the bolts are apt to work loose and it does not brace the parts. The construction now used is that of gusset plates riveted to the posts and girts. The wind rods were also connected directly to the posts at the girts. The bolt holes, as originally inserted through the post angles, weakened the posts since they reduced the net section. The rods are now connected to the gusset plates. The arrangement of these parts is shown in Fig. 5. The diameter of bolt and thickness of plate are proportioned to provide proper bearing strength.

22 The posts and girts of steel towers erected to support steel tanks, and to some extent wood tanks, are now largely made of channels latticed on both sides or having a plate on one side. Other shapes such as the Bethlehem H-beam and two channels with an I-beam between to form an H-section are also used to some extent.

23 Competition in the manufacture of these structures has resulted in the use of too high unit stresses and as a result the posts, figured on a conservative basis as represented in case of other structural work such as bridges, had a factor of safety of less than 4 and sometimes as low as 2½. Failure has resulted, an example of which is shown in Fig. 7, a view of a large structure after a collapse. To obtain safe towers it became necessary, therefore, to set maximum allowable stresses. The loading of the structure consists of the weight of the

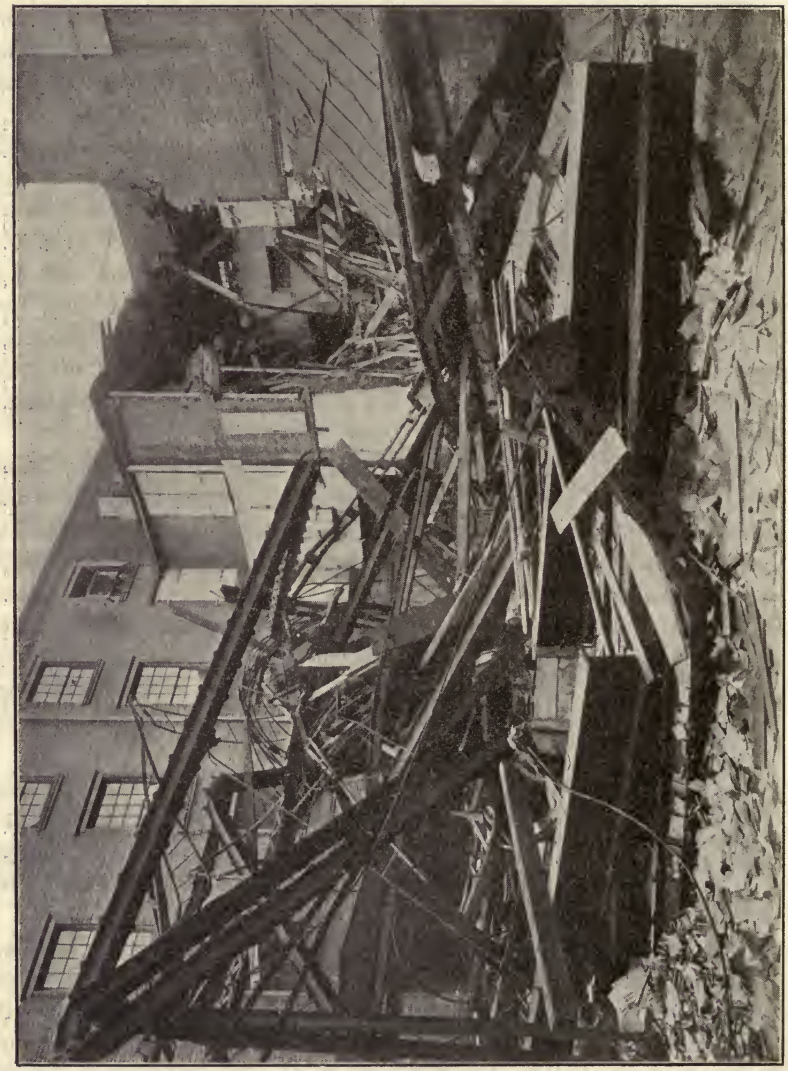

FIG. 7 VIEW OF COLLAPSE OF STEEL TOWER STRUCTURE WITH LARGE WOODEN TANK AND DAMAGE CAUSED THEREBY

structural and ornamental steel work, platforms, roof, piping, etc. The live load consists of the weight of the total volume of water; the movable load on the platform is assumed to be 30 lb. per sq. ft. and the wind load. The wind pressure is assumed at 30 lb. per sq. ft. and that on the tank is this pressure times 6/10 the projected area of tank and roof, and in the case of steel tanks, the curved bottom. The total wind pressure on the posts, struts, wind rods, ladders and riser boxing is assumed at 200 lb. per linear ft. of height of tower.

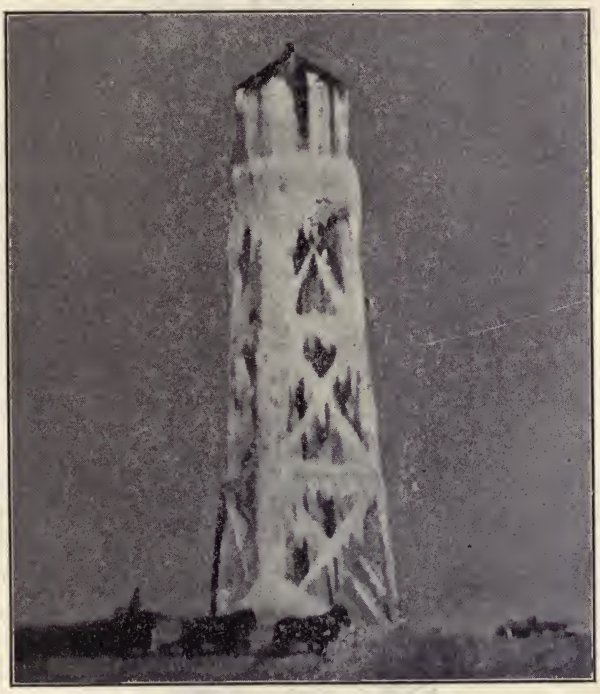

FIG. 8 UNUSUAL CASE OF LOADING OF LARGE STEEL TOWER STRUCTURE BY ICE; 60,000-GAL. TANK ON A 79-FT. TOWER, UNINJURED AFTER THAW

24 An unusual loading of a steel tower structure by ice is shown in Fig. 8. The tank is 60,000 gal. on 79-ft. tower and is located at Richfield, Idaho. The ice coating was formed from repeated overflowing of the tank during cold weather. When warm weather melted the ice, the structure was found to be uninjured. This experience illustrates the abuse a well-built structure will withstand.

25 All parts of the structure are proportioned so that the sum of the dead and live loads shall not cause the stresses to exceed those

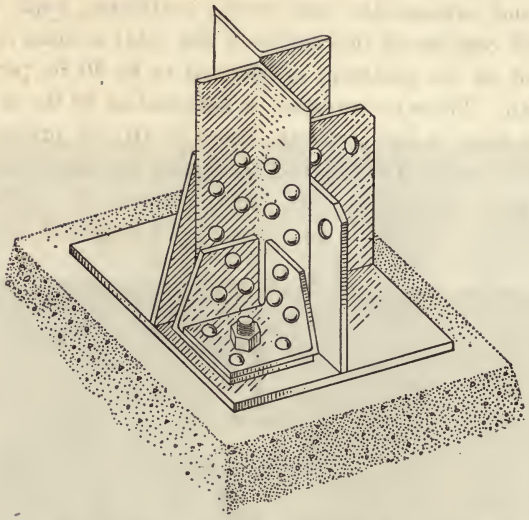

FIG. 9 TYPICAL CONSTRUCTION OF FOOTING FOR ANGLE IRON COLUMN

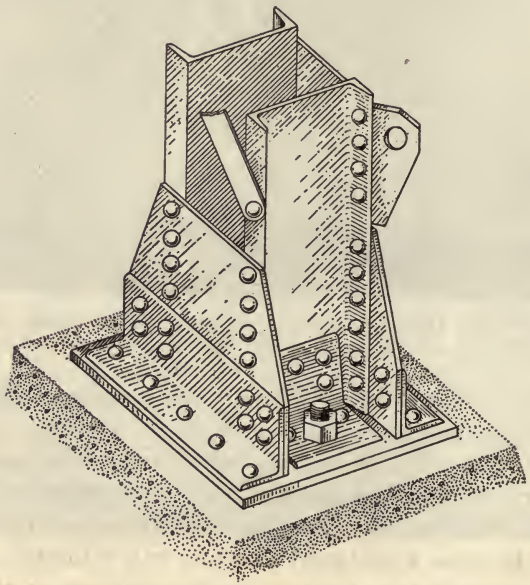

FIG. 10 TYPICAL CONSTRUCTION OF FOOTING FOR CHANNEL COLUMN

allowable. The principal stresses in such a tower structure are axial compression on gross section of columns and struts, axial tension on net section of wind rods, bending on extreme fibers or net section of rolled shapes, built sections and struts, and shearing of rivets. The axial compression on gross section of columns and struts is determined from the following expression: $17{,}100 - 57\frac{L}{R}$, where L is the unsupported length of the member from center to center of connections in inches, and R the least radius of gyration in inches; the ratio $\frac{L}{R}$ should not exceed 125 for columns and 150 for struts and minor members and the maximum compression allowable as thus determined is 12,000 lb. per sq. in. The axial tension on net section of wind rods must not exceed 12,500 lb. per sq. in.; the bending on extreme fibers or net section of rolled shapes, built sections and struts 16,000 lb. per sq. in., and the shearing for shop driven rivets, 10,000 lb. per sq. in. and field driven rivets 7,500 lb. per sq. in.

26 The lower ends of the posts have not been as carefully designed as their importance requires. Frequently, in angle iron towers particularly, no special attempt has been made to properly distribute the load to the base plate attached to the post footing. Cast-iron plates were first used and the concentrated loading caused these to crack, resulting in collapse or in throwing the structure dangerously out of plumb with possibility of failure of the foundation under this post. The present use of steel plates has improved conditions, but the design must be such as to distribute the load to them as shown in Figs. 9 and 10, which are designs that are being used quite generally.

27 In anchoring the columns to the foundations, the diameter of the bolt at root of thread should be such as to withstand the maximum uplift due to the wind with tank empty, and to resist the shearing force at base plate. The bolts should be made from round wrought iron or mild steel rods without upsets.

FOUNDATIONS AND SUPPORTS

28 The foundation piers to support steel towers are usually made of concrete, consisting of one part portland cement, three parts clean sand and five parts broken stone. They are usually pyramidical in shape and proportioned to suit soil conditions. The allowable bearing pressures on soil will range from 1 to 5 tons per sq. ft., depending on the quality of the soil. Where the soil is moist or rather loose, a

girt should be provided at the base of the tower to prevent spreading of the posts. The allowable bearing pressures for footings should not exceed 400 lb. per sq. in. for portland cement concrete and 200 lb. per sq. in. for ordinary brick work with portland cement mortar, except when the tank is to be rested on the building walls, when the bearing plate should be figured on the basis of 125 lb. per sq. in.

29 The weight of the foundation pier when buried at least two-thirds of its height should be equivalent to the calculated net uplift due to wind pressure with the tank empty, that will be transmitted to it; otherwise it should be one and one-half times that amount.

30 Where the tank structure is above a building, and the building walls are depended upon to act as supports, great care should be taken to determine that the construction is safe against collapse. In many cases, tanks are supported by building walls not originally built to carry them, but where a sprinkler system was later installed it was considered more convenient and cheaper to use the walls than to erect a detached tower for the tank. This has frequently been done without making a thorough inspection first of the condition of the walls, and, largely through ignorance, the necessary care was not taken to distribute the load. Many failures have consequently resulted and there are no doubt numerous cases of this kind where the tanks are apt to fall at any time. A serious cracking of the walls in such an installation is shown in Fig. 11.

31 Inspection should be made of the quality and condition of the brick and mortar or other material used in the construction. The wall foundations should be examined as to construction and bearing on soil or rock. The condition of the bond between abutting walls should be noted and a general inspection made for sizable cracks in the walls. The thickness of walls and size and spacing of window and door openings should be measured and calculations then made to determine if the load of tank, water and trestle can be safely distributed over the walls. All unnecessary openings should be bricked or otherwise solidly filled in, and it may be necessary to sacrifice some openings to obtain the required strength. When the walls cannot be altered to support the load, the additional support required can be obtained by carrying steel beams down inside the walls to a solid foundation, provided these do not interfere with the occupation of or the processes carried on in the building. Otherwise it will be necessary to provide a separate steel tower.

32 The proper strength of foundations is especially important because of the greater probability of loss of life from the falling of a tank from above a building as compared with the falling of a tank on a detached tower. The monetary loss is liable, of course, to be also much greater, as the water will undoubtedly wreck the building and cause heavy water damage. The building departments of cities endeavor to obtain proper construction, but unfortunately they do not

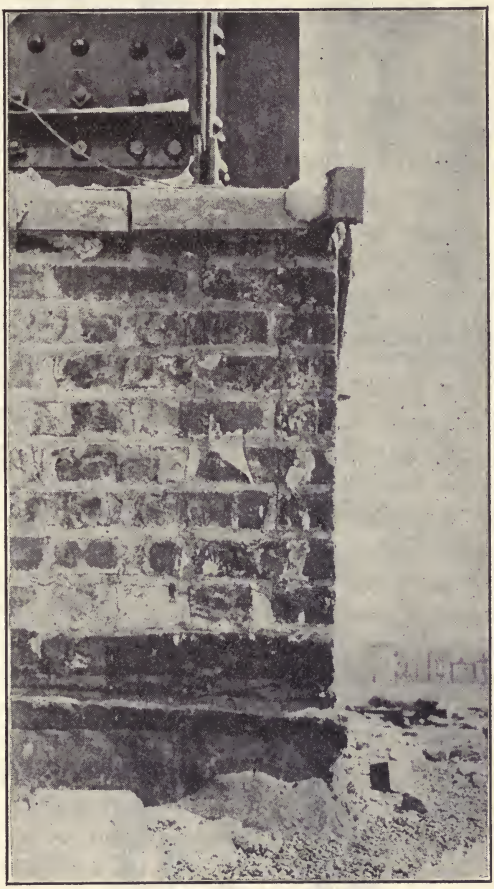

FIG. 11 VIEW OF SUPPORTING WALL THAT HAS CRACKED UNDER WEIGHT OF TANK

always succeed. The possibility of trouble is increased because of the divided responsibility of the tank builder and the architect. The former seldom concerns himself as to the strength of the supporting

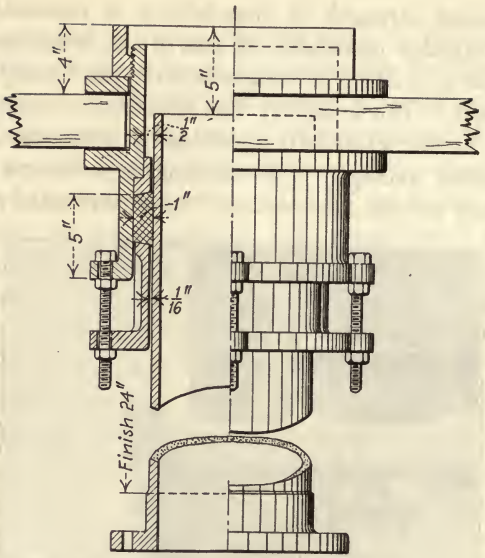

FIG. 12 EXAMPLE OF PROPER DESIGN OF EXPANSION JOINT FOR WOODEN TANK

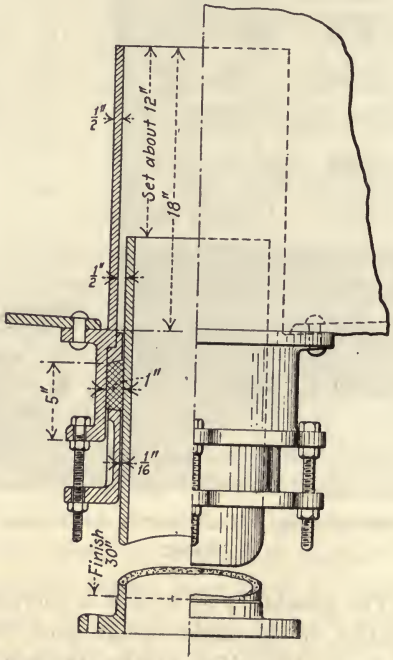

FIG. 13 EXAMPLE OF PROPER DESIGN OF EXPANSION JOINT FOR STEEL TANK

walls, assuming that the latter has given the matter proper attention, so he goes ahead and erects the tank according to contract.

GENERAL FEATURES

33 Tank fittings should receive careful attention to insure the reliability of the equipment. The discharge or riser pipe is more serviceable if made up of cast or wrought-iron pipe, flanged or coupled, than one made up of bell and spigot pipe, since the latter is apt to leak at the leaded connections, necessitating removal of the frost-proof boxing to permit of repairs. A tank and tower is constantly swaying from side to side and this tends to loosen up leaded joints. Furthermore, the increased rigidity of the flanged and coupled pipe permits the use of a minimum number of tie rods. There are usually four rods connected, one to each post, at girt connections.

34 The connection of the discharge or riser pipe to wooden tanks has usually been made by extending the pipe through ordinary cast-iron slip flanges bolted to the tank bottom on each side of the opening. The hole in the planks was cut larger than the size of the pipe to form a packing space which was filled when parts were first assembled. A better construction was used for steel tanks having a stuffing box and gland. Both types of joints were found to be unserviceable, however, the former because the joint could not be tightened when leakage occurred, and the latter principally because the iron to iron parts rusted together, which resulted in the breaking of some pipe fitting and the emptying of the tank. Examples of properly designed expansion joints forming tank connections for wooden and steel tanks are shown in Figs. 12 and 13, respectively. These have a bronze gland and ample clearance between the iron parts to prevent binding by corrosion. The packing space is large and the joint is extended within the tank bottom to form a settling basin, to prevent sediment getting into the yard pipe and clogging the sprinklers at time of fire.

35 A tightly constructed frost boxing should be placed around the discharge or riser pipe, and arrangements made for keeping the water heated by a hot water heater or a steam coil in the bottom of the tank. Designs of three-ply, two air-space boxings are shown in Figs. 14 and 15.

36 A tank level indicator or telltale is necessary to give a positive indication that the tank is full at all times. After many serious fires it has been learned that the tank had been partially or wholly empty at the start of the fire, and the lack of water had handicapped the fire protection devices. Tanks may be left empty due to neglect, but

usually so because of false indication of the telltale. The most used type of device for this purpose is the float in the tank water, operating a target sliding on a scale fixed to the outside of tank. Obviously,

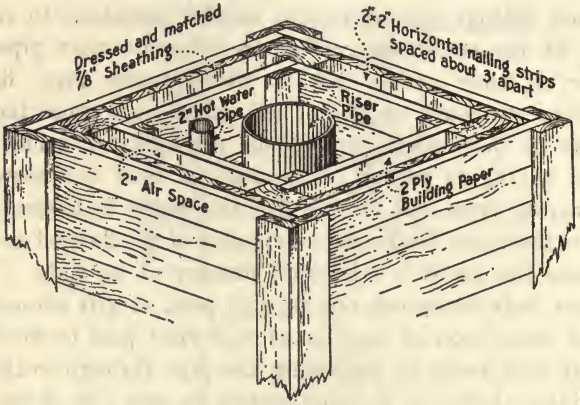

FIG. 14 DESIGN OF EFFICIENT FROST-PROOF SQUARE BOXING FOR ENCLOSURE OF RISER PIPE TO TANK CONNECTION

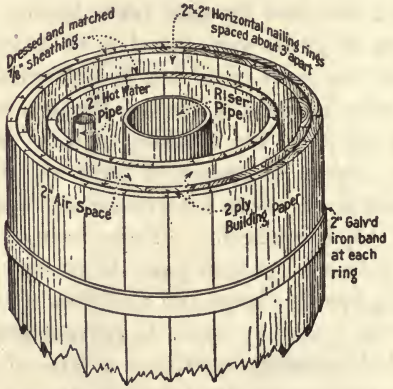

FIG. 15 DESIGN OF EFFICIENT FROST-PROOF ROUND BOXING FOR ENCLOSURE OF RISER PIPE TO TANK CONNECTION

these are subject to sticking due to their mechanical construction and exposure to snow and ice in freezing weather. The ordinary pressure gage has been largely used but cannot be positively depended on, since it is seldom, if ever, tested and the parts stick, causing false readings. There are several types of electrical telltales operated by a float, but these are complicated and easily gotten out of adjustment. Attention is also necessary to maintain the electrical current.

37 The most reliable telltale is undoubtedly the mercury gage, an adaptation of which for this purpose is shown in Fig. 16. This gage was developed by the laboratories of the Associated Factory Mutual Fire Insurance Companies. It should be placed indoors where it will be observed and cared for. The mercury pot is then piped to the

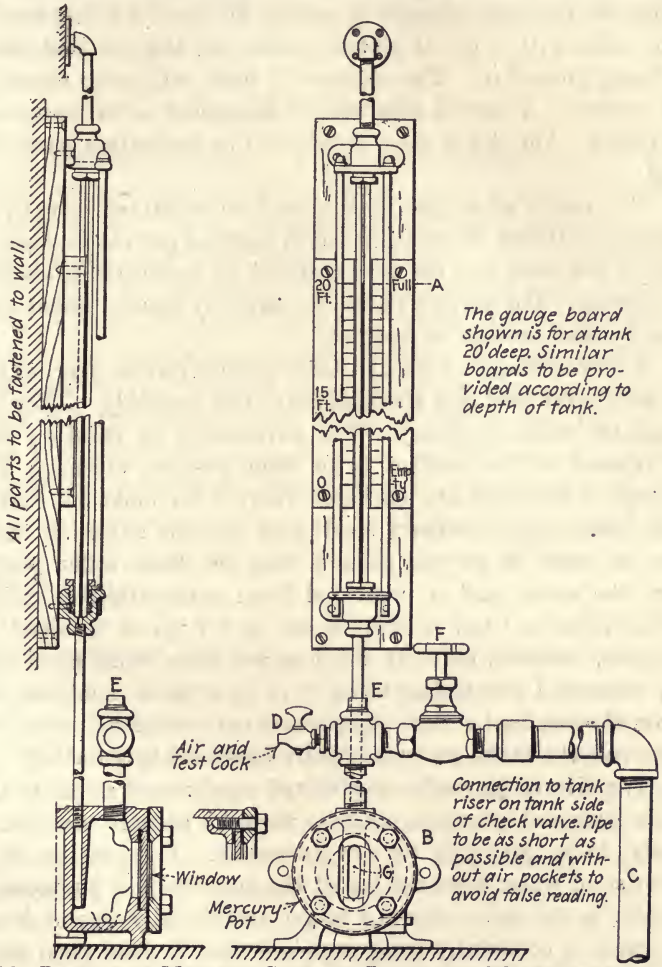
All parts to be fastened to wall
20 Ft.
Full
A
15 Ft.
0
Empty
The gauge board shown is for a tank 20' deep. Similar boards to be provided according to depth of tank.
F
E
E
D
Air and Test Cock
Connection to tank riser on tank side of check valve. Pipe to be as short as possible and without air pockets to avoid false reading.
B
C
Window
Mercury Pot

FIG. 16 DETAILS OF MERCURY GAGE FOR INDICATING WATER LEVEL IN TANK

riser pipe on the tank side of the check valve, and the gage board adjusted after filling the mercury pot. The gage is readily tested by opening the pet cock on the water pipe. If water continues to flow under constant pressure, the apparatus is in operative condition; otherwise, the pipe is clogged or there is a valve closed.

38 The painting of steel tanks and towers and of the iron hoops of wooden tanks is very important to prevent corrosion. Steel plates and shapes should be given the usual priming coat at the shop. The surface of the metal should be thoroughly cleaned of mill scale, rust and grease and be perfectly dry before applying the paint. A good paint for the first coat is made by mixing 20 lb. of red lead and 10 lb. of zinc oxide with 3 qt. of boiled linseed oil, the red lead and zinc oxide being ground in. This amount of paint will cover about 50 sq. yd. of surface. A second coat should be applied after structure has been erected. For this a more durable oil or asphaltum paint should be used.

39 The inside of a steel tank should be repainted, usually every two years, or oftener, if the paint shows signs of peeling or wear. The outside of the tank and the tower should be repainted at about five-year intervals. The surface should be carefully cleaned either by sand blast or by steel brushes or scrapers.

40 The iron hoops of wooden tanks should receive a priming coat as for structural steel and a second coat after assembly. They should be repainted when necessary. The advisability of painting wooden tanks exposed to the weather is an open question although a large percentage of the tanks are painted. There is no doubt but that paint protects wood under ordinary conditions, but the objection raised to its use on tanks is on the ground that the tank water percolates through the staves and is prevented from evaporating as it is held under the paint and this is likely to set up dry rot in the wood. It is well known, however, that dry rot does not occur when wood is completely immersed but rather when it is in a moist condition in the presence of some heat. This objection is not considered well-founded and as a rule the tanks are undoubtedly preserved by painting.

41 The life of properly constructed equipments depends largely upon the care and attention given to them by property owners. The tanks should be used only for fire protection. The practice of using a foot or so of water from the top of the tank for mill purposes is objectionable as the tank collects a larger amount of sediment from the water which is constantly being supplied than it does when used for fire service only. This sediment is likely to settle in the sprinkler pipes and either clog them completely, or, if the sprinklers are open, seriously interfere with their discharge. If water is drawn from the bottom for mill purposes the tank may be empty when needed for fire service. Furthermore, the fluctuation in water level is apt to result in shrinkage of the upper ends of the staves of wooden tanks, causing

leakage and hastening corrosion in the steel tank by the repeated wetting and exposure of the sides to the air.

42 Manufacturers in general have awakened to the advisability of following the best practice in the designing and construction of tanks and towers as represented by the foregoing, and are now making structures which are serviceable and safe.

DISCUSSION

J. W. KETLER[1] (written). Improperly designed supports for tank structures on buildings and the sub-foundations are likely to endanger life and property. As an engineer making a specialty of this work I come in contact with faulty designs, principally in the building itself. The fault does not lie entirely with the structural engineer, but in a great many cases with the architect. I would suggest stringent rules governing the installation of these outfits, especially the thorough examination of the building by a competent engineer or architect familiar with this class of work. I would also suggest, and would co-operate to further the acceptance of one uniform set of specifications governing the engineering manufacture and erection of towers and tanks, both on the ground and on buildings, by all insurance companies, and so far as possible by individuals, corporations or city building departments.

B. A. FREEMAN[2] (written). Lack of uniformity in the design of tanks and towers at present is well known to those interested in that line of work. Besides the cause suggested, close competition, the lack of ability on the part of most purchasers to discern the merits and demerits of designs submitted helps not a little towards that end. There is little novel or unusual in the construction of these structures. All of the problems connected with them have been solved before in connection with other engineering works. Now that elevated tanks and towers have come into general use for fire protection purposes, it is advisable both to protect the purchaser and the bidder who wishes to erect a good structure, that a specification be forthcoming which shall place all competitors on the same basis. A worthy attempt in that direction has been recently made by the engineering department of the Associated Factory Mutual Fire Insurance Company of Boston,

[1] Chief Engineer, Wondnagel & Co., Chicago, Ill.

[2] Engineer, The Rusling Company, New York.

and intelligent inspections in connection with such a specification should produce very good results.

BRYAN BLACKBURN[1] (written). Too much credit can not be given to Mr. Teague and his associates for the splendid service rendered along the line of tank betterment. The specifications issued by his office, if honestly carried out, will afford a high-class structure that is simple in design, economical in construction and very efficient in service. I have designed a vast number of elevated tanks for fire protection, and I can say without fear of contradiction that the tanks built under the requirements of Mr. Teague's office are beyond criticism.

I differ with Mr. Teague, however, in that it is allowable to single-rivet the vertical seams above the first course. As pointed out in my article on this subject in The Engineering Magazine,[2] the vertical seams should be double riveted, not so much for strength of joint, as to prevent the breaking of the calking edge by breathing tendency of shell due to change in water level; also it is my opinion, based on experience, that the riveting should be closer than is theoretically required for efficiency of joint, in order to insure that the plates be well drawn together under field riveting. The joints in the main column at strut points should be milled to insure full bearing.

The formulae and stresses set forth in the paper for main members are correct and ample, but I would suggest that, while the struts should be of such section as to provide for all live and dead loads, and that the depth of strut should be such that the unit stress due to weight of member should not exceed 4000 lb. per sq. in., still my experience indicates that more often the sizes of these struts are fixed by the requirement that no strut shall exceed 150 radii of gyration in length.

In the using of the Bethlehem H column for main columns, as suggested by Mr. Teague, care should be exercised to see that the column lengths do not exceed 125 radii of gyration. The standard practice in these elevated tanks allows very long column lengths in the latticed channels columns and this has led some inexperienced designers to employ H shapes in too long lengths, with results that are not pleasing, and in several cases that have come under my observation, partial failures have resulted, not from any inherent defect of the H sections but from misuse.

[1] Assistant Engineer, R. & D. Cole Mfg. Co., Newnan, Ga.

[2] Elevated Tanks for Fire-Protective Service, p. 385.

Stress must be laid on the fact that the so-called balcony used on hemispherical-bottom steel tanks is not an ornament but a horizontal girder subjected to large loads due to the horizontal component of the stress in the column.

A. H. HAYES (written). It is generally conceded that in all structures on which the lives of men, or the safety of property from destruction are dependent, too much care can not be taken in design and construction; this is the business of the designing engineer. It is also his duty in this day of efficiency, so to plan his structure that he may use no more material than is necessary to take care of the maximum loading which may occur, with, of course, the proper factor of safety.

In the latter part of the section on towers, the author states that "the total wind pressure on the posts, struts, wind rods, ladders and riser boxing is assumed at 200 lb. per lineal ft. of height of tower." This is taken to mean that for all towers carrying tanks, whether of 10, 20, or 100,000 gal. capacity, the wind pressure must be assumed to be the same. This appears inconsistent, inasmuch as the size and shape of the members will vary with the style of construction, and the loads they are to carry.

The writer has, in several years' experience, designed many towers to carry the smaller tanks on which the wind pressure, taken at 30 lb. per sq. ft. of flat surface, did not reach 150 lb. per vertical ft.; also others of larger capacity, where the wind pressure greatly exceeded 200 lb. per vertical ft. This is a small consideration when designing low towers, but with comparatively high towers where the wind stresses at the base of the tower equal or exceed those caused by the weight of tank, tower and water, the difference amounts in some instances to 15 per cent. Therefore, a strict economy of material is not possible.

I wish to present for consideration another point along this line not brought out in the author's paper, but which is embodied in the Specifications for Gravity Tanks and Towers, recently published by the Associated Factory Mutual Fire Insurance Companies, with which specifications, it appears, the author is largely to be credited; that is the practice of assuming that the wind pressure on the tower is the same when acting in the direction of the diagonal of the tower as when acting in the plane of the bents. This is not correct when the posts are made up, as with many of the smaller towers, of two angles

placed corner to corner forming a star section as illustrated in Fig. 4. Here the wind surface, when the wind is acting in the direction of the diagonal of the tower is $\frac{S}{\sqrt{2}}$ Therefore, the maximum compression stress in the post on the leeward side of the tower, due to wind on the posts, is exactly the same as when the wind is acting in the plane of the bents. The pressure in the struts, wind rods and riser boxing is the same whether the wind acts in one direction or another, and the practice of increasing these stresses by the square

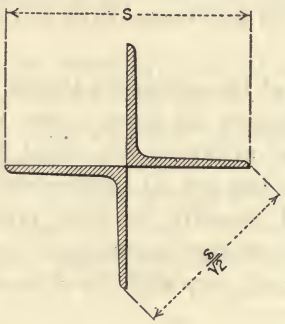

FIG. 17 SURFACE EXPOSED TO WIND IN POST FORMED OF TWO ANGLES

root of 2 is correct. But in some towers the wind pressure on the posts is the greater part of the total. Take a case where 50 per cent of the total wind pressure is on the posts; then instead of increasing the total wind stresses by the square root of 2, only 50 per cent should be thus increased.

Taking into consideration the fact that an assumption of 200 lb. per vertical foot in many cases raises the total actual stresses in the posts 15 per cent, and that the latter assumption increases them another 10 per cent, it appears that the author has gone even beyond the limits of conservatism, and in order to meet these requirements the manufacturer is compelled to use considerably more material than is needed. This matter should be left as an incentive to each individual engineer to design his structure so as to present the least possible wind surface, and thus produce an economical design.

ALBERT BLAUVELT (written). I take issue with the statement recommending a water level indicator to be placed indoors. Such an

indicator is limited to showing the water level in the tank and may not do that because the operator can not be relied upon to use the test cock alluded to, to prove that the connections are clear. As a matter of field service, it is preferable to use a sight overflow pipe placed vertically inside the tank, with the upper end at the desired full elevation of water and the lower end extending through the bottom of the tank and angled over for easy seeing. By such an indicator, much more is accomplished. The operative condition of the means for filling the tank is proved, the operator can not guess that the tank is full enough; he must fill the tank to get an indication, he must look at the tank structure and thereby is more or less compelled to inspect it, and if the heat has failed and the tank frozen, the top of the sight pipe freezes shut and the tank gives warning of ice by overflowing around the top of the tank instead of at the sight pipe.

G. A. SMITH[1] (written). The paper is principally an abstract of the author's specifications covering gravity tanks and towers prepared for the Associated Factory Mutual Fire Insurance Companies of Boston. There are a few points in them with which I do not fully agree, but on the whole consider that the specifications cover the field in very good shape. There are two points, however, which I might mention and on which I would like to have the discussion of different members of the Society:

In the section on Towers for Elevated Tanks, Mr. Teague gives a formula for unit stress in water tower columns. This formula gives a unit stress of 12,000 lb. on the basis of 90 radii. It has always been my contention that a higher unit stress is permissible in the design of water tower columns for the reason that the entire load is quiescent and is entirely dead load. There might be some question raised on this point due to the fact that a certain portion of the column load is caused by wind pressure; however, it is the general practice to permit a higher unit stress for wind loads than for either dead or live loads.

It is a general practice of most engineers in the design of steel structures, such as bridges and buildings, to use a higher unit stress for dead loads than for live loads. For instance, Theodore Cooper specifies for bridges a unit stress in chord compression members of 15,000 lb. for dead load and approximately 7000 lb. per sq. in. for live load, both of these stresses being based on 90 radii. On a similar

[1] Manager, Des Moines Office, Des Moines Bridge & Iron Co., Des Moines, Ia.

basis, I believe that a unit stress of 15,000 lb. per sq. in. for water tower columns is conservative. I would mention, however, that on account of there being so much sentiment against this high unit stress, the company which I represent has adopted a unit stress of 12,000 lb. per sq. in. for columns in their standard water tower specifications.

Another point on which I do not quite agree with Mr. Teague is the expansion joint illustrated in his paper. The illustration is not quite clear as to the kind of metal used in the different parts, but it is his requirement to use cast iron in the pipe sleeve and the collar, and brass or bronze in the follower and adjustment bolts. I contend that this expansion joint will not permit proper operation after it has been used for some time. It is my idea that the cast-iron sleeve will corrode or rust more or less above and below the gasket, and in fact will corrode to some extent on the surface covered by the gasket. After this sleeve has become more or less corroded, I believe that it will not slide freely through the gasket, and in fact there is considerable danger that the riser pipe would probably give in some other point before sliding in the expansion joint as intended. I cannot see why the brass follower gland used by Mr. Teague is any better than a cast-iron gland for the reason that the clearance between the sleeve and the follower is sufficient to prevent any serious adhesion by corrosion.

My idea of an expansion joint which would always work freely, would be to have a brass covered sleeve instead of cast iron so that the surface coming in contact with the gasket would not be affected by corrosion and therefore would slide freely through the gasket.

C. S. Pillsbury[1] (written). Mr. Teague has done a great deal to bring about the general adoption of proper methods in the design and construction of sprinkler tanks, and his specifications are generally recognized by the manufacturers as fairly representing the best present practice in that class of work. In this paper, he outlines a number of suggestions as to proper unit stresses, plate thicknesses, etc., all of which tend to result in a safe and economical structure. The provisions which prohibit bolted connections and reduce the shear on field rivets are especially good.

As Mr. Teague says, it is of great importance to design the details so they will develop the full strength of the main members. To those familiar with railroad bridge shop practice, this statement will seem

[1] Chicago Bridge & Iron Works, Chicago, Ill.

superfluous, but, as a matter of fact, practically every failure of an elevated tank can be traced to an eccentric top post connection, insufficient provision for the horizontal thrust at the top of the posts, poorly made column splices, or some other oversight due to inexperience. To prevent excessive stresses in the tank shell, the number of posts must be proportioned to the diameter and depth of the tank, and provision must be made to resist the torsional moment due to the curvature of the sides. It is also necessary to provide for a number of severe forces other than gravity and wind loads. This last statement is well emphasized by the illustration of a water tower loaded down with ice due to the continuous overflowing of the tank in cold weather. It is only a carefully designed structure that can undergo such treatment without injury.

Following are stress formulae used by the Chicago Bridge & Iron Works which may be of interest in connection with Mr. Teague's paper:

Tank Sides and Bottom. The stress in pounds per linear inch in the sides of a cylindrical tank is $2.6 \times H \times D$, where D = the diameter of the tank in feet and H = the head of water in feet. The maximum stress in a hemispherical bottom is $1.3 \times H \times D$ and in an elliptical bottom $2.3 \times H \times D$. The last two formulae are closely approximate, H and D being the same as before, except that in this case H should be taken as the total depth of the tank.

Posts. The vertical component of the dead load post stress is equal to the total water and metal load divided by the number of posts. The vertical component of the wind stress equals the following:

3-post tower ..... $\frac{M}{0.75\,D}$

4-post tower ..... $\frac{M}{1.00\,D}$

6-post tower ..... $\frac{M}{1.50\,D}$

8-post tower ..... $\frac{M}{2.00\,D}$

where M = moment of wind about panel point at the bottom of the post section considered and D = diagonal of tower at the panel point about which moments are taken.

Rods. There is no dead load stress in the rods. The wind stress equals the following:

3-post tower	0.500 (V-V') Sec. A
4-post tower	0.707 (V-V') Sec. A
6-post tower	1.000 (V-V') Sec. A
8-post tower	1.307 (V-V') Sec. A

where V' = vertical component of post stress in panel above;
V = vertical component of post stress in same panel as the rod,
and A = angle rod makes with the vertical.

Struts. Except where the batter of the posts changes there is no dead load stress in the struts. The wind stress is approximately the horizontal component of the rod stress in the panel below.

THE AUTHOR. As stated in the paper and as emphasized by Mr. Ketler, the possibility of danger to life and property from elevated tanks is greatest when they are erected above buildings. I fully agree with Mr. Ketler that buildings should be carefully examined by a competent engineer to ensure that they will safely carry the load to be imposed upon them. His suggestion that a standard specification should be adopted by all parties interested is a timely one, as the recently developed specifications of the Associated Factory Mutual Fire Insurance Companies can be used as a basis, such changes or additions being made as seem necessary so that the standard will thoroughly cover all conditions. It would be desirable to have this Society assume the leadership in the work.

Mr. Freeman also advocates uniformity in the design and construction of tanks and towers, and recommends a standard specification following along the lines of those of the Factory Mutual Companies.

In reply to Mr. Blackburn's suggestion that the vertical seams above the first course be double riveted, I have not considered it necessary to advise this, since experience with the single-riveted joints has thus far been quite satisfactory. Certainly if much trouble from leakage at these joints should be experienced, the double riveting would be advisable.

Mr. Hayes has brought up a point which should receive more careful treatment, i.e., the wind load on the tower members and pipe fittings. I have followed current practice in assuming the total wind pressure on these members as 200 lb. per linear foot height of tower and have not, as yet, attempted to vary the pressure to suit various

heights and types of towers, although I believe that this is advisable, especially in the case of towers to support tanks for other uses, such as water works' supplies. The 200 lb. pressure gives reasonably accurate loadings for sprinkler tank towers, as the height of these is fairly constant within limits. A closer approximation of the loading, even for these towers, would, however, be appreciated by the designing engineer.

The further point which Mr. Hayes mentions, that the actual wind pressure varies according as to whether the wind blows at right angles to plane of the bent or diagonally is, of course, correct, especially for angle-iron towers. As a further refinement in determining the wind load on the structure, it would be well to give this matter consideration.

Mr. Blauvelt's comments regarding the use of the overflow to determine if the tank is full, represent the best practice, and I would say that the use of a mercury gage as a telltale is intended merely as a constant indication of the water level, in place of the unreliable float and target telltale and other devices.

In reply to Mr. Smith, 12,000 lb. per sq. in. is considered a proper maximum allowable unit stress in compression for tank towers by most manufacturers and engineers. This stress is none too conservative, in view of the very special type of the structure where failure of one post member results in immediate and complete collapse. Furthermore, the towers are exposed to the weather, and as they are usually the property of comparatively small companies unfamiliar with the proper care of steel structures, they will not be inspected and kept painted as will steel railroad bridges and other structures, and so the steel members should have some additional thickness, in order that they may remain serviceable even after some considerable corrosion has taken place.

I agree with Mr. Smith that a brass or brass covered pipe at the expansion joint is preferable to the iron pipe and believe that this would remove the last possibility of binding through corrosion in the present design. This improvement may be secured in time, but the expansion joint shown is so much better than what has been used that it seemed wise not to add further to its cost at this time, since there undoubtedly would have been serious objections raised by the manufacturers for commercial reasons. The advantage of the brass follower is, of course, its non-corrodibility, which will permit its being moved to tighten the packing even if the iron pipe is corroded, as

mentioned by Mr. Smith. The oil lubricant with which the packing is saturated will naturally prevent any appreciable corrosion of the pipe where it bears on it and so permit of movement of the pipe without injury to the packing.

The suggestions made by Mr. Pillsbury are important and the data which he has given regarding stress formulae will undoubtedly be useful.

No. 1424

FIRE PUMPS

By Ezra E. Clark,[1] Boston, Mass.

Non-Member

Were we to review historically the development of fire pumps, we would find a long list of efforts, dotted frequently with failures, and marked here and there with a partial success. No sooner does man meet fairly well the fire pump requirements of an age than he finds he must make a still better apparatus; for in the march of human progress, the needs of fire protection always keep just a little ahead of the provision and the vigilance of man. We could find much of interest in this buried history. We would find how the masterpiece of some old-time mechanic had been the cherished marvel of a community and had been forced by a rival into a state of obsolescence. Even the fire pump of today will sometime become an obsolete device. Nevertheless, it represents the present state of the art. It is the pump on which we depend to stiffen the pressure of a weak public water supply. It is the pump from which we expect prompt and reliable service in the extinguishing of fires. It is the business man's concern to know something about these pumps, their possibilities, their record, which one of the various types should be selected for a given situation, and how to install them so as to secure their best service. Are not these vital questions for the man who expects to spend a limited appropriation for fire protection? He may ask, "Why do I need a fire pump when I have a tank on my mill tower and a 6-in. connection to the public main?"

2 An answer to his question may be found in the history of the Paterson, N. J., fire, eleven years ago, where a group of six mills with their own fire pump protection, checked completely the conflagration sweeping towards them. Each mill had its own fire pump and a stored water supply, the supplies being supplemented in some cases by a connection to the public mains. So intelligently were these moderate fire equipments handled that when the fire's defeat was made

[1]Engineer and Inspector, Factory Mutual Fire Ins. Cos.

Presented at the Annual Meeting 1913, of The American Society of Mechanical Engineers.

certain, there was still water to spare in the pump cisterns, and the mills had sustained no material damage. In another instance, quite recent, where there was no pump but simply a tank and a moderate public water supply, the fire ran so rapidly through the building that 50 sprinkler heads opened, and several fire streams were quickly brought into service. These combined drafts on the water supply

FIG. 1 VIEW OF FIRE STREAMS FROM CENTRIFUGAL PUMPS

so reduced the pressure that neither sprinklers nor fire streams were at all effective, and the building was destroyed with a loss of $8000, the value of several fire pumps. There are very many instances that could be cited where the prompt starting of a fire pump has assured ample pressure, a good distribution of water from the sprinklers or several effective fire streams, and a comparatively small loss.

3 The argument is being advanced today that because 90 per cent of our fires are being extinguished with perhaps a few sprinkler heads, rarely requiring the service of a fire pump, insurance interests are demanding supplies of water far more generous than is reasonable, and that much smaller supplies and smaller pipe connections would amply suffice to give reasonable protection. This logic is attractive to the man who pays for the equipment. But fires have a way of proceeding along illogical lines and it is the wise man who holds himself

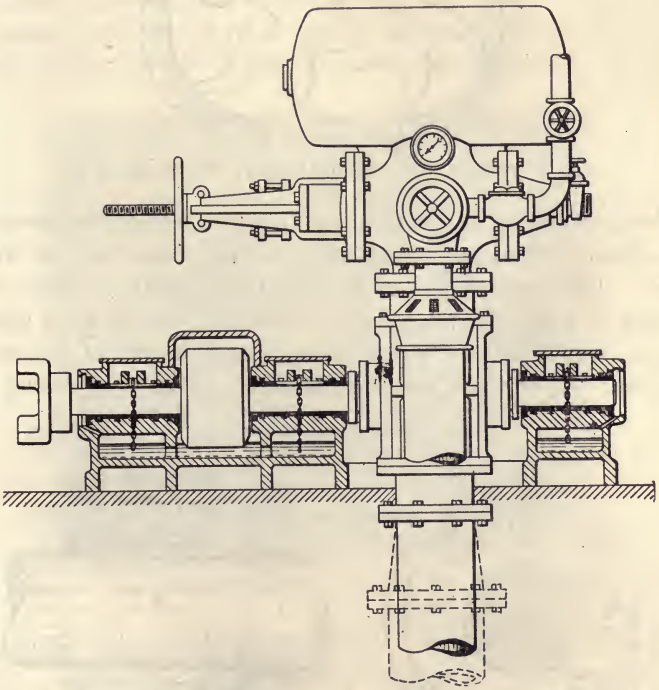

FIG. 2 PART SECTION OF ROTARY PUMP, TYPE B

prepared for the illogical and the unexpected condition, even for the worst conceivable condition, the doubtful 10 per cent.

DEFINITION OF TERM

4 Broadly speaking, any device by which water can be thrown on a fire is a fire pump. In its simplest form, it is a bucket of water and a man. In the earliest days the bucket was supplemented by huge syringes in the form of a bladder or bellows, the gut of an ox serving as a length of hose. Later they were made from brass cylinders and

were called "squirts." From these crude appliances were developed the hand and steam fire engine around which cluster the memories of many exciting contests during the last century. These devices, however, are all portable in character, and valuable time is often lost in bringing them to the scene of action.

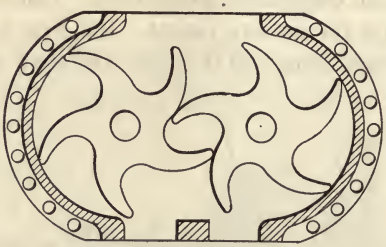

FIG. 3 WORKING CAMS OF ROTARY PUMP, TYPE B

5 Manufacturing properties, particularly, were often located at such a distance from fire engine stations that prompt service was impossible. The incalculable value of a few gallons of water at the beginning of a fire became self-evident, so that pumps fixed upon the premises, connected to a water supply and a source of power, were introduced. Later these were connected to a system of piping about

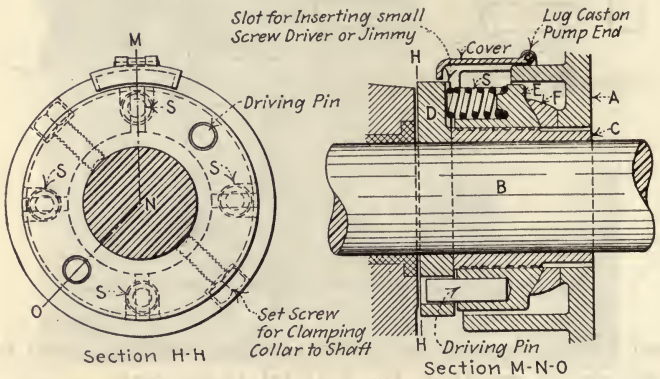

FIG. 4 WATER JOINT USED ON ROTARY PUMP SHAFTS. TYPE B, AS SUBSTITUTE FOR STUFFING BOX

the yard and in the mill, and the pump then became available to supply sprinkler heads, or furnish fire streams from hydrants.

6 In this paper, the term fire pump is to be used in its restricted sense, i. e., a pump that is installed in a fixed position for fire purposes. In this sense there are three or four distinct types, generally acknowl-

edged: rotary, duplex, centrifugal, and power pumps, any one of which, of course, could serve as a fire engine by mounting it on wheels and providing a source of power.

TYPE OF PUMP

7 The type of pump to be selected should be determined by the character of power available for the purpose. Naturally the most reliable source of power at any situation would be chosen, and then a pump selected that can best be driven with that power. This should be the general rule. Every type of pump, however, has its own limitations and these must be kept in mind when studying the whole

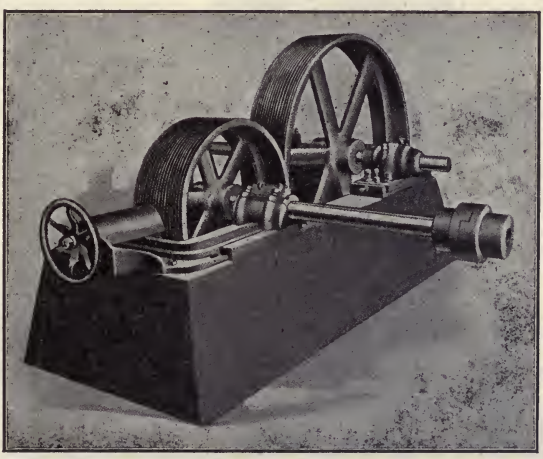

FIG. 5 TYPICAL PRESS FRAME FOR FRICTION GEARS DRIVING ROTARY PUMPS

problem. The natural selection would be a pump that can run at or near the speed of the prime mover. One instance has been noted where a centrifugal pump was connected to a waterwheel through a pair of spur gears having a ratio of 5 to 1, and another case where an electric motor was connected to a rotary pump through spur gearing having a ratio of 1 to 5. In each case, the pumps took their suction supply under a head. Both cases may be dismissed as examples of poor engineering, for the requirements of fire service would have been much better served if the two pumps were to exchange places and the reduction gears omitted, thus securing the simplest, cheapest and most reliable arrangement.

8 The earlier mills were operated almost exclusively by waterwheels and their usual speed made it comparatively easy to connect

with rotary pumps. Thus, a great many pumps of this type found their way into the wheel pits of mills, although they were too often, unfortunately, poorly located. Power plunger pumps were used in a few cases, but the rotary came to be the prevailing pump, owing to its simplicity, cheapness and adaptability to the prevailing power. Where waterwheels of ample power and moderate speed are available, the rotary type of pump becomes the logical choice, for the two can often be direct-connected, or some simple form of transmission, such as

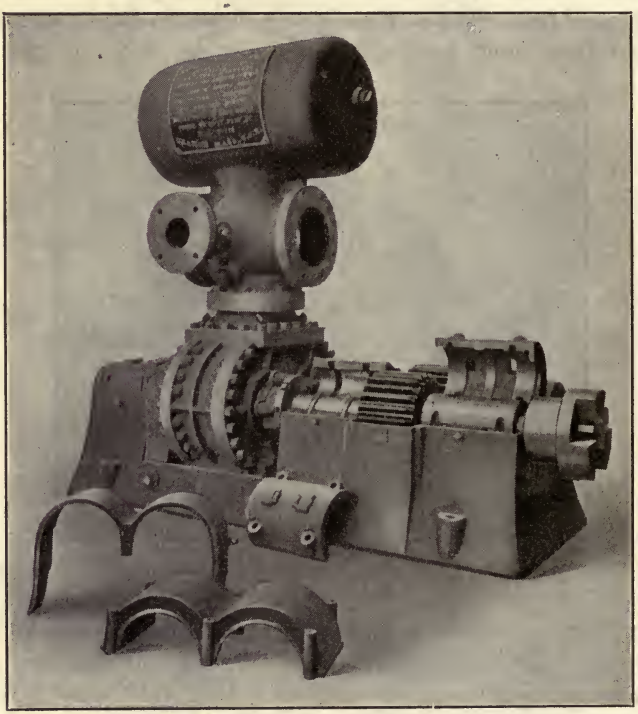

FIG. 6 ROTARY PUMP, TYPE B, PARTLY DISMANTLED

friction gears, or a silent chain having a low speed ratio can be employed. The impulse form of waterwheel with its high speed would find a better selection in the centrifugal pump, requiring no intermediate gearing. In weighing the claims of the several pumps for a situation, three things should be kept in mind: the character of the power available, the limitations of the pump and the fact that the simplest drive is the best.

ROTARY PUMPS

9 Among the earlier types of pumps installed in mills for fire purposes was the rotary. There are so many indifferent rotary pumps in mills and factories that the mere mention of a rotary raises doubts as to its value for fire purposes. Still, it should be remembered that even with the imperfect construction of the old time cast-iron rotary, they have given some very effective service at critical times. However, I invite your attention not to these obsolete pumps, but to that product of the rotary pump builder that meets the present insurance specifi-

FIG. 7 ROTARY PUMP, TYPE B, WITH STANDARD FITTINGS

cations and notably to that which is designated as Type *B* (Figs. 2 and 3).

10 Several features of this design were proposed and developed by the author in 1904. They were not immediately adopted, as some experimental work was necessary, and new patterns were required. These new features were chiefly as follows: In place of the usual two sets of overhung cast-iron gears, one extra-heavy pair of cut steel gears is substituted, preferably forged solid with their shafts; these gears are supported on either side by generous bearings, and the two shafts are larger and stiffer; all bearings are provided with liberal

oil reservoirs and ring or chain oilers; in place of the usual stuffing boxes, a special form of metal water joint has been adopted, which is giving excellent service, thereby avoiding the use of perishable packing. The usual fire pump features are also provided; viz., the casing and working parts exposed to corrosion are of bronze; there are more liberal water passages; the working cams are of a new design to insure reasonably smooth running; and the usual air chamber, hose connections, an approved spring relief valve and starting valve complete the arrangement. The press frame carrying the friction gear is now fitted with a heavy spring that maintains the gears in contact,

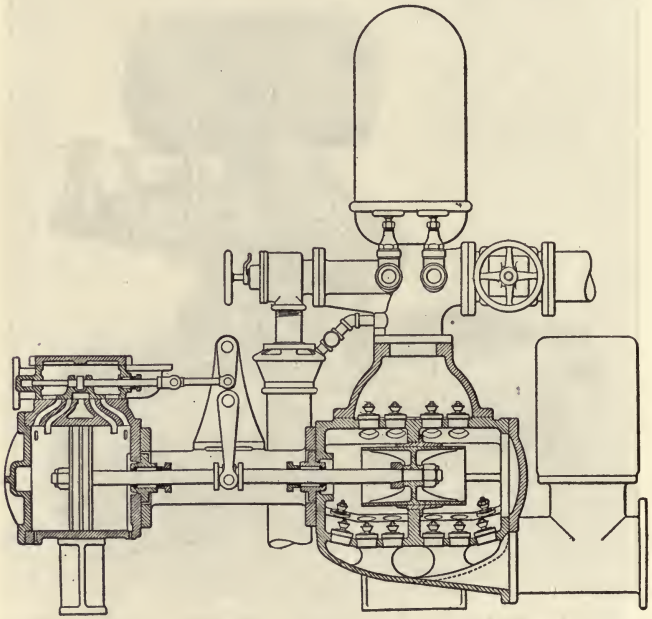

FIG. 8 SECTIONAL VIEW OF UNDERWRITER STEAM FIRE PUMP

and compensates for a possible lack of truth in their running (Fig. 5).

11 These pumps are not intended for daily service as the wear between cams and casing would soon render them inefficient. But for the brief periods of fire service, considerations of wear are of less importance than certainty of action. Full particulars as to their construction can best be gathered by a study of published specifications. The effort has been to secure a rugged, reliable pump capable of doing hard work, resisting corrosion and withstanding abuse. These pumps are obtainable today from at least two manufacturers, and cost

somewhat less than a duplex pump. The limited number that are already in the field are so far giving excellent satisfaction.

DUPLEX PUMPS

12 Where steam is the most reliable source of power, the duplex "Underwriter" pump is usually the choice, because of its record of proved value. It is also much the cheapest of any steam-driven fire pump of acceptable reliability, selling for approximately ten cents a pound. In spite of its numerous parts, liable to derangement, this pump has been so thoroughly boiled down, so to speak, that with present day construction it is proving to be one of the most reliable of fire protection appliances. The steam-turbine driven centrifugal pump is, of course, available for those who are willing to pay the price, but it costs approximately twice as much. Its main advantage over the duplex is its simplicity of working parts. Its disadvantages lie in the fact that a pressure of 100 to 125 lb. of steam is required for good results, whereas, with a duplex pump, 50 lb. of steam is ample for ordinary fire pressures. The outfit is also subject to the same limitations as to suction lift as all centrifugal pumps. When the steam turbine centrifugal has become standardized and its price made more nearly competitive with that of the duplex, no doubt it will receive a much wider introduction.

13 As steam began to supplement and in cases to supplant water power in mills, steam-driven pumps were installed to help out or replace the rotary. At first the single cylinder pump was used, but the advent of the duplex pump marked the beginning of a better machine for fire purposes. The duplex type of fire pump is so well known that it is hardly necessary to describe it, full information as to construction being available in published specifications. When the development of the Underwriter type of duplex pump was undertaken by Mr. John R. Freeman in the early nineties, he found the pump as made for the trade lacking in several features needed for fire protection. The frequent failures in the field mainly showed that improvement was needed. But pump builders did not fully sense the importance of some of the features that were demanded, and more or less opposition was encountered.

14 The old trade duplex pumps were found with steam and water passages restricted, but these have been enlarged in the Underwriter type, and a higher speed thus made possible. The trade pumps were iron fitted and became rusted through neglect, so that they could not be started; the Underwriter pump is brass fitted wherever

corrosion is liable to seize the working parts. In the older pumps, cast-bronze valve stems were used that worked loose or broke off; in the present Underwriter standard, these stems are of rolled bronze and made heavier, and the guard is so secured in place by a nut lock devised by the author that the parts cannot work loose (see Fig. 9). In the old trade pump it was common practice to provide means of adjustment for the valve motion, and this adjustment sometimes worked loose, or was tampered with, putting the pump in a lame condition; in the Underwriter design, all adjustment has been ruled out and a simpler and more reliable mechanism is now used. Special fire service fittings are attached, such as priming valves, starting valve, relief valve, hose valves, oil pump and lubricator, all these parts having been subjected to examination and test before adoption. (See Figs. 8 and 10.)

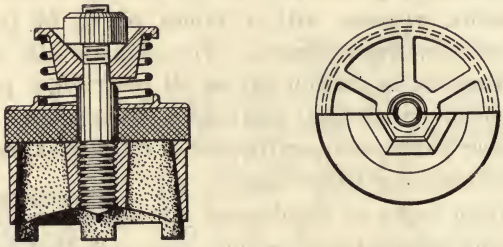

FIG. 9 DETAIL OF VALVE STEM AND SPECIAL NUT LOCK

15 To incorporate these changes has taken time and it has been more or less an up-hill job. Experience, however, has shown the value of these improvements and has indicated from time to time still further improvements which have been adopted, so that today we have a fire pump that is able to start at a moment's notice, run at full speed if needed, and when supplied with steam, water and oil, will run continuously and deliver its rated capacity, or more, against a fire pressure of 75, 100, 150, or even 200 lb. in special cases.

16 A fire pump to meet these conditions must be rust-proofed. It must be built of material chosen for strength, toughness and durability. It must be so free from complications as to permit of operation by men of moderate ability. It must be able to withstand safely a large measure of abuse, and be, so far as possible, "foolproof." This is the sort of steam pump that is being furnished for fire protection today. It is not claimed that the pump is built like the wonderful

"one hoss shay," equally strong in every part, but rather from specifications that make the possibility of failure extremely remote.

ELECTRICALLY DRIVEN PUMPS

17 Where the electric motor is the most reliable source of power, the centrifugal pump naturally becomes the choice. The speed of a

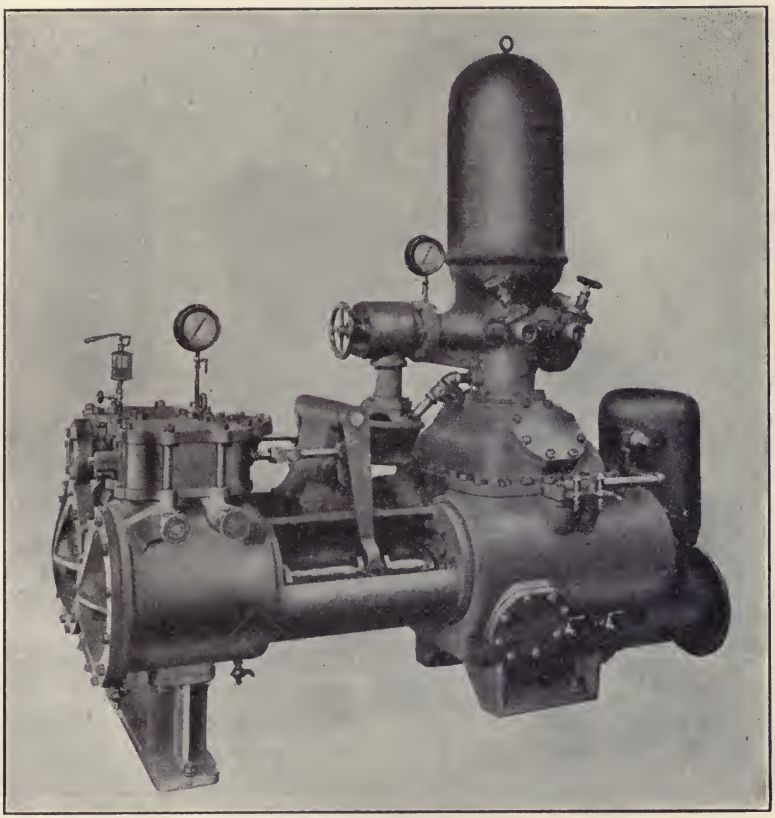

FIG. 10 GENERAL VIEW OF UNDERWRITER STEAM FIRE PUMP

centrifugal pump can readily be made to conform to the speed of the standard motor and all intermediate gearing avoided. The direct-connected motor and pump thus form the simplest possible arrangement. It has, however, certain limitations which will be noted later, and there may exist conditions which some form of power pump electrically driven would meet with better satisfaction.

18 Comparatively few power pumps are in use as fire pumps, owing to their highest cost. Their best field is where a daily service pump of good efficiency is needed, that can in the emergency be used for fire purposes. The triplex type of power pump is much to be preferred to the single or duplex forms, owing to its steadier discharge pressure, and as it is usually built in the vertical form the floor space required is the minimum. A strong point in favor of the triplex power

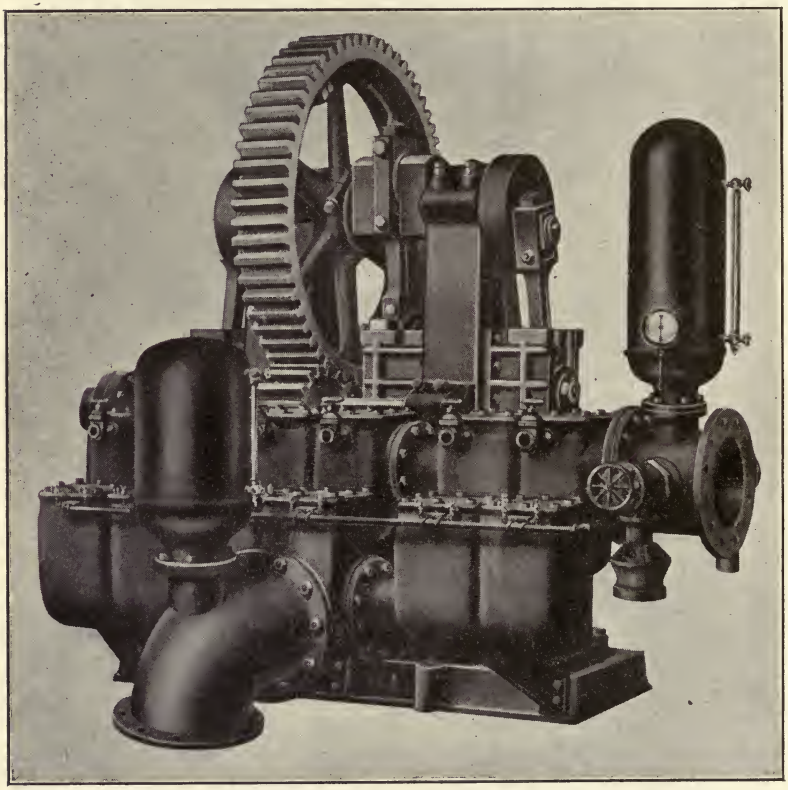

FIG. 11 VIEW OF TRIPLEX POWER FIRE PUMP

pump as compared with a centrifugal is its higher efficiency, running as high as 80 to 85 per cent for outside packed plungers as against the usual 65 to 70 per cent for the centrifugal. That type of triplex pump employing one centrally located driving gear is much to be preferred to those having two overhung gears. Experience has shown that the central gear drive will better ensure an even distribution of power, and thus avoid undesirable stresses on gear teeth and possible

breakage. A notable example of the central gear triplex power pump may be found in the high-pressure fire service station in Philadelphia, where seven such pumps are operated by gas engines, and have given excellent satisfaction with a minimum of repairs.

19 The centrifugal pump is comparatively a newcomer in the fire pump field. It is distinguished from all others in being a *velocity* pump, the pressure varying directly as the square of the peripheral velocity of the impeller. The electric motor is no doubt responsible for its coming so rapidly into favor. The constant speed motor direct-connected to the pump makes the simplest and cheapest arrangement and affords a fairly wide range of capacities and pressures. The variable speed motor or steam turbine, costing a little more, would place the pump more nearly on a par with the steam duplex as to range of pressures, and, the author believes, would make the preferable

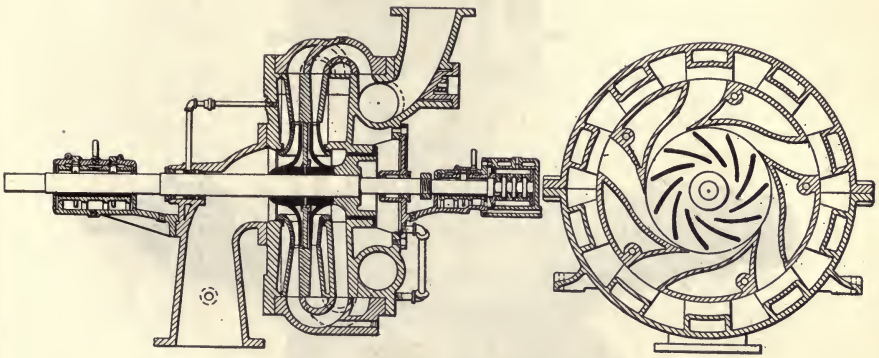

FIG. 12 CENTRIFUGAL FIRE PUMP WITH DIFFUSION VANES

outfit. Its greater cost, and complication is the general objection to it.

20 There are two types of centrifugal fire pumps built, one with the simple cylindrical case, known as the volute pump, and the other with a diffusion vane casting or chute case. The diffusion vane type, sometimes called the turbine pump, employs a series of gradually enlarging passages that permit a more gradual change of the velocity of the water into pressure than does the volute, thus resulting in some slight gain in efficiency. This, however, is not an essential part of an acceptable fire pump, and as the volute type of pump (Fig. 13) is much simpler in construction, and yields efficiencies that are satisfactory (60 to 70 per cent), this type of centrifugal fire pump is being generally adopted. This type also lends itself better to

the horizontal division of casing, which is preferred to the circumferential division, as it permits quicker and better access to the interior for overhauling or cleaning.

21 The simplicity of the centrifugal, having but one moving part, appeals to every engineer. There are no valves to choke up, no plunger to wear out, no valve motion or gears to break, and no dangerous pressures possible even with all outlets closed, unless specially provided with a variable-speed prime mover. The discharge is steady, smooth, devoid of shocks, and more nearly approximates that from a gravity supply than any other pump. But it has its limitations, which should

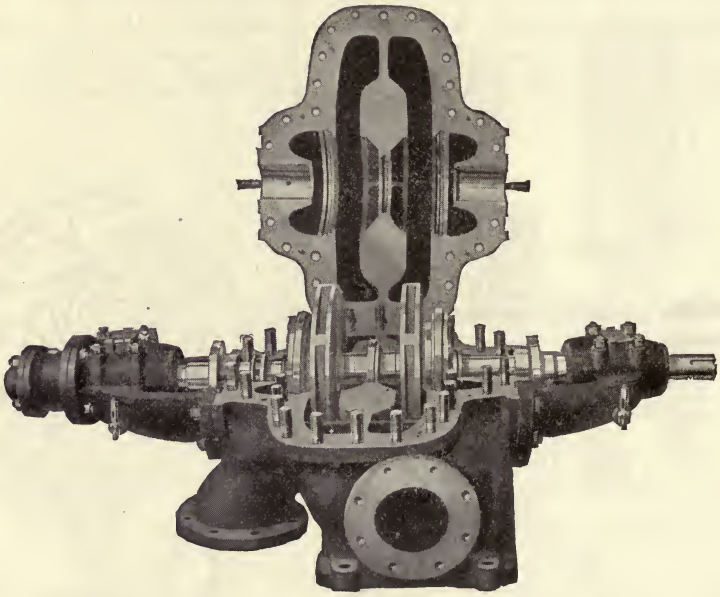

FIG. 13 VOLUTE TYPE OF CENTRIFUGAL FIRE PUMP

be fully realized and considered in making an installation. Unless provision is specially made for a variable speed, its maximum working pressure is usually not planned to be very much above 100-lb. pressure, running for the smaller flows in some cases up to 130 or 140 lb. Consequently the water pressure needs of a situation should be carefully studied before deciding on this form of pumping outfit, and if higher pressures are needed, either a special form of impeller should be substituted or a variable speed provided.

22 A centrifugal has no power to exhaust the air from a suction pipe, as has the rotary, duplex, or other displacement pump. For this

reason, the suction supply should come to the pump under enough head to flood the pump casing, thus insuring its being primed. If a lift is unavoidable, then it becomes necessary to provide a suitable foot valve and a generous supply of priming water, enough to fill completely the suction pipe and pump casing. This feature, of course, limits the centrifugal to situations where there are easy suction supply conditions.

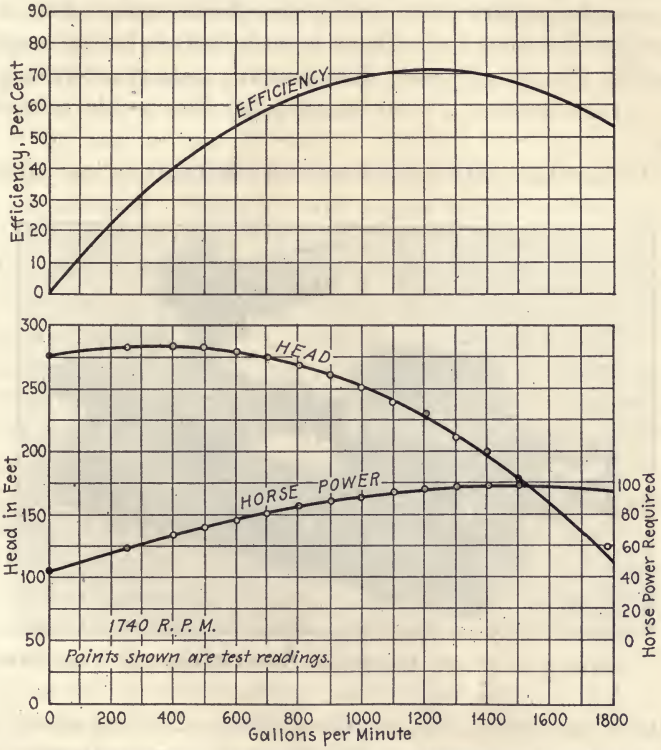

FIG. 14 CHARACTERISTIC CURVE FROM TEST OF 1000-GAL. CENTRIFUGAL FIRE PUMP

23 Fig. 14 shows a characteristic curve obtained by actual test from a 1000-gal. centrifugal fire pump. It will be noted that at its rated capacity of 1000 gal., its discharge pressure runs about 108 lb., and at half capacity, or 500 gal., it runs as high as 123 lb., while at 1500 gal. the pressure drops only to 78 lb. Thus with a constant speed motor it is possible to get a good range in discharge capacities, 50 per cent above and 50 per cent below the rating, and all at pressures that would be serviceable for fighting fire.

24 The curve shown is approximately what is desired and obtained in the average two-stage centrifugal. For some situations it may be desirable to secure a different form of characteristic, and this curve can be made steeper or flatter within limits.

25 Where two pumps are to operate together these curves should be similar and the characteristic curves should be known and posted in view of the operator, in order that he may intelligently operate each pump to perform their fair share of the work. The shut-off pressure (no discharge) should not be materially below the maximum pressure, at the top of curve. For pumping against a constant head, a curve approximating a horizontal straight line would naturally be chosen.

26 In cost the centrifugal pump with bedplate is but little more

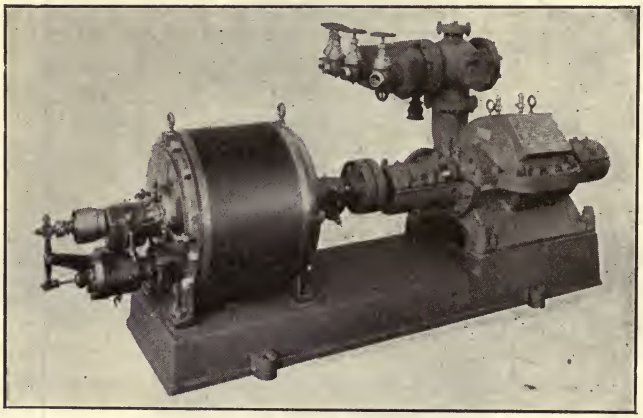

FIG. 15 EXAMPLE OF STEAM TURBINE DRIVEN CENTRIFUGAL FIRE PUMP UNIT

than the duplex per pound, but the addition of driving motor brings up the price very materially, so that the outfit combined costs 15 to 16 cents per lb., and weighs about the same as a duplex pump. For a given type, the speed of a motor determines its size and influences the total price of outfit. It may roughly be stated that the centrifugal pump and motor costs 50 per cent more than the duplex. There is a wide difference, however, in the weights of the same size pump among different manufacturers.

INSTALLATION

27 In a brief paper like this, the details of a pump's installation cannot be properly reviewed, and yet the success or failure of a fire

pump depends very much on its being properly erected and connected up. Briefly, it may be said that a fire pump must be so located as to be safe at all times from the breakage of its pipe connections due to the falling of floors and machinery, and safeguarded from any influence such as smoke, fire or flood that might drive away the operator. It should be accessible at all times for operating and overhauling, and not blockaded by miscellaneous storage. It must permit of practicable pipe connections, the suction pipe receiving first consideration, and the steam and discharge pipes so run as to be safe from damage and always be in commission. The exhaust should go direct to atmosphere, and not be tied up with other pipes.

28 The suction supply should preferably be practically inexhaustible, such as a river or lake, and a suitable intake, properly screened, provided. Where a stored supply only is available, it should be large enough to supply the pump for two hours, more or less, depending on conditions. No cast-iron rules can well be established as the insurance engineer having jurisdiction is expected to weigh conditions, and secure a reliable pump service without expensive or complicated refinements. Full details as to pump installation have been covered in the several publications and specifications which from time to time have been developed in the Factory Mutual Inspection Department.

29 It will perhaps be of interest, and not altogether without value, to look a bit into the future and try to discern what is to be the development of the fire pump. It is not at all unlikely that the steam turbine centrifugal will gradually displace the duplex pump, as its design becomes simplified and standardized, and its cost lowered. Where steam is not available, we shall find the electric-driven centrifugal, or the gasolene-driven rotary, and as fast as its development permits, the gas-turbine-driven centrifugal will possibly win recognition and receive adoption. To whatever extent insurance interests adopt these new appliances, it will be done with an eye single to their proved value, as regards simple and rugged construction and reliability of performance.

DISCUSSION

Alfred B. Carhart. Upon the practical side of the operation of fire pumps, I think the author of this paper has called attention, in one of the final paragraphs on Installation, to a very important consideration, which was not mentioned in the presentation of the paper.

It is such a simple matter, that it seems surprising that engineers in laying out new plants, will connect the exhaust of steam-operated fire pumps with other exhaust lines, and then supply stop valves by which these pipes can be cut off from the exhaust line to prevent back-pressure and condensation, not realizing that when the fire pump is needed in a hurry, it will be impossible to operate it effectively, because there will be no free outlet for the exhaust steam. Such conditions have been discovered, much to the chagrin of those in charge of otherwise admirably constructed plants.

Albert Blauvelt (written). Referring to the section of Mr. Clark's paper on electrically driven pumps, it is my observation in the field that the electric motor centrifugal pump is as yet considerably short of working development as compared to the Underwriters' steam fire pump. The steam pump is a self-contained affair including the more essential fittings, and ordinarily is handled successfully when needed. The electrically driven pump thus far appears always to require considerably more skill and sense of time on the part of the operator. The motors are excellent and so are the pumps, but neither pump maker nor motor maker appears to know each other, nor do either appear prepared to deliver a complete pumping set, fully fitted and adapted to unfamiliar and incompetent handling in a degree comparable to the steam pump. Meantime, mismoves, or delay or total failure of starting, mark the attempt of watchmen or other men not trained to the electric pump. I take no pride in the various electric pumps installed in our practice thus far, and have less esteem for similar jobs set up under other auspices. Doubtless as the trade increases some shop will put out a complete and self-contained electric pump, with fittings interlocked or timed against mismoves.

The Author. From the remarks of Mr. Blauvelt, it appears that he finds that men not trained to the electric pump make mismoves, or totally fail to start the pump. This is also true of men not trained to steam or rotary pumps.

It should be remembered that the steam fire pump has been growing up with us for 40 or 50 years, and there is a larger number of average men familiar with such apparatus than there is familiar with the electrically driven centrifugal pump. The latter has been with us but a few years, and for the average man it is still somewhat of a novelty that he but vaguely comprehends.

A remedy for this situation does not appear to be getting the pump maker to know better the motor maker. It is rather for the pump operator to know better the apparatus under his charge. It has not been found necessary for rotary pump builders to furnish the water-wheel driving the pump, nor has it been necessary for steam pump makers to go into the steam boiler business.

The troubles arising from centrifugal outfits are mainly due to errors in installation, or a failure to realize the limitations as well as the possibilities of this type of pump.

A properly installed centrifugal and motor is one of the simplest of fire pumps, and easy to start. Regarding interlocking devices to prevent mismoves, these are but complications that add to the confusion of the operator. A simple device under intelligent control is much to be preferred to automatic control involving added complications. It is not necessary for a man to be an electrician in order to operate a motor driven centrifugal, any more than it is necessary for a steam pump operator to be an expert in hydraulics, pneumatics, and the use of steam. But he does need to know something about these matters in either case, and especially to know the use of the fittings and devices that he sees before him. There is no positively certain way to prevent an incompetent man from making the fool move.

There will no doubt be improvements made from time to time based on experience tending to standardize these outfits, which will simplify their operation; but the lack of confidence in present apparatus which is expressed by the gentleman is not shared by the author.

No. 1442

FLOOR SURFACES IN FIREPROOF BUILDINGS

By Sanford E. Thompson, Boston, Mass.

Member of the Society

In fireproof construction, whether it be office building, factory, or institution, the question of the type of floor surface to select and the method of construction to adopt is a most important one. The constant tread and shuffling of feet cause a friction that it is difficult to withstand without serious wear.

2 From the construction standpoint, in a non-combustible structure a cement surface is in keeping with the rest of the building and is naturally the first considered. In many instances the cement concrete or granolithic floor has proved extremely satisfactory, while in others, because of the use of improper constituents, of inexpert construction, or of its selection for places to which it is not adapted, it has proved a disappointment. As a matter of fact, no one type of floor surface is adapted to all conditions, while for any type that is properly selected, the choosing of the materials and the manner of the construction will govern to a large extent the durability of the surface.

3 It is the purpose of this paper to discuss briefly the different kinds of floor surfaces, and to compare their various qualities, their cost, and their adaptability to specific conditions. This is followed by a more detailed treatment of the methods of constructing the concrete or granolithic surface which have produced satisfactory results.

4 An engineer in consulting practice is called upon frequently not only to design and construct but to investigate defective construction and also to make special tests for the determination of the best methods to employ in a particular case. In this paper are embraced not only the results of experience in floor construction and repairs, especially as they relate to granolithic surfaces, but also the

Presented at the Annual Meeting, December 1914, of The American Society of Mechanical Engineers.

conclusions derived from special tests and investigations made in connection with services as consultant on the superstructure of the New Technology buildings in Cambridge.

5 Embodied in the paper is much of the material forming a report to the Stone & Webster Engineering Corporation, engineers and builders of New Technology. The matter covered, then, will include:

Discussion of selection of type of floor surface
Relative costs of various floor surfaces
Characteristics of floor surfaces
Tests and investigations of granolithic construction
Recommendations for granolithic construction

SELECTION OF TYPE OF FLOOR

6 The selection of the type of floor is dependent on the character of the structure, the nature of the wear, and the architectural appearance. Every building must be considered by itself. Suggestions for the type of surface to select are covered in the following pages. As a preliminary guide, the material suitable for different conditions may be given as:

Basements: Granolithic finish with trowelled surface made with approved materials and workmanship.

Factory Floors: Granolithic finish with trowelled surface; hardwood.

Machine Shops: Granolithic finish with trowelled surface; hardwood on substantial base.

Ground Floors for Heavy Manufacturing: Wood block; granolithic.

Warehouses: Granolithic with trowelled surface; asphalt composition; hardwood.

Offices: Hardwood; linoleum on concrete; magnesium composition.

Corridors and Halls for institutions and office buildings: Terrazzo; granolithic finish with ground surface.

Entrance Pavilions: Terazzo; mosaic; tile; natural stone.

Class Rooms, Lecture Rooms, and Drawing Rooms: Linoleum on concrete; granolithic with ground surface; hardwood; magnesium composition.

Laboratories: Granolithic with trowelled surface; magnesium composition; tile; hardwood.

Lavatories: Terrazzo; granolithic finish with ground surface; tile.

The above selections are given in the order in which choice might be made for the average building or room of each class.

APPROXIMATE COST OF VARIOUS FLOOR SURFACES

7 In giving approximate costs of floors it is recognized that the condition of the market both in labor and materials and the quality

of the materials selected, affects the unit price to a very large degree; also the location, size and shape of the rooms to be finished.

8 The following prices are based on estimates of cost in place. For the materials like hardwood that are laid after the partitions are placed, the prices apply more particularly to a building such as a college or other institution divided into offices and rooms of various sizes. Each price is assumed to include total cost of the labor and material, exclusive of the structure itself. It is assumed that the base upon which the floor is laid is either structural concrete or some similar material.

	Cost per sq. ft.
Granolithic:	
If laid at same time as base, with trowelled surface	$0.05
If laid at same time as base, with ground surface	0.08
If laid after completion of base, and trowelled	0.07
If laid after completion of base, and ground	0.10
Linoleum:	
Battleship linoleum including $0.03 per sq. ft. for placing and trowelling a ¾-in. layer of mortar immediately after base concrete is laid, linoleum being figured at $1.30 per sq. yd., cemented in place	0.18
Hardwood:	
Maple or birch, single thickness, including $0.01 for leveling off base concrete and including stringers with cinders between, with rough 1-in. floor underneath	0.22
Maple or birch, single thickness, including $0.01 for leveling off base concrete and including stringers with cinders between	0.18
Maple or birch, single thickness, including $0.01 for leveling off base concrete and including stringers with cinders between, with rough 2-in. floor underneath	0.25
These prices are based on good quality of hard wood at about $45 per 1000 ft. B. M.	
Terrazzo:	
With small stone ⅛ in. to ½ in. including $0.01 for leveling the base concrete	0.19
With large stone ⅜ in. to 1 in. including $0.01 for leveling the base concrete	0.24
For areas of 50,000 sq. ft. or more, deduct 10 per cent from these figures.	
Base 6 in. high	0.36–0.50 per lin. ft.
Marble Mosaic:	
Grouted and ground	0.50–0.60
Magnesium Composition:	
For large areas, say 100,000 sq. ft., including $0.01 for leveling base concrete	0.20
For small areas, say 25,000 sq. ft., including $0.01 for leveling base concrete	0.24

Asphalt Flooring:
Including $0.01 for leveling base concrete..... 0.15–0.17
Asphalt Mastic:
For areas 100,000 sq. ft. or more including $0.01 for leveling base concrete..... 0.15–0.16
In chemical laboratories..... 0.17–0.18
6-in. sanitary base,..... 0.25 per lin. ft.
Tile:
Quarry tile..... 0.35–0.40
Fancy pattern tile..... 0.50
Cork Tile:
Moravian tile, fancy pattern..... 0.75–1.25

CHARACTERISTICS OF FLOOR SURFACES

9 *Granolithic Trowelled.* As ordinarily laid in buildings, granolithic or concrete surfaces are subject to dusting and under heavy traffic, such as trucking, are liable to serious wear. On the other hand, experience with first-class construction and tests of actual floors shows that it is possible, by proper selection of the aggregates and expert workmanship, to reduce the dusting to an insignificant amount and to produce a surface hard enough to stand even severe wear.

10 For factory floors, notwithstanding many cases of inferior construction, the use of granolithic is largely increasing. It is becoming recognized that the durability of granolithic is in a very large measure dependent upon the sand or other aggregates used in the construction and the methods of laying it.

11 The chief objection to concrete or granolithic surfaces for offices, drafting rooms, class rooms, and certain laboratories, is that it is dull in appearance, hard on the feet for men standing all day, tends to break tools dropped upon it, and is not adapted to attaching seats and other furniture readily, especially where they have to be shifted occasionally. In certain colleges, however, concrete surfaces are used widely and highly recommended. At Bowdoin and at the University of Wisconsin it is considered satisfactory for all purposes. At the University of Missouri the newer buildings are all being built with granolithic surfaces. In some colleges granolithic is being satisfactorily used for corridors. Most of the colleges favor granolithic for chemical, mining, and mechanical laboratories. The Leland Stanford, Jr., University states that in the mechanical and engineering laboratories the men complain of hardness and coldness, requiring wood platforms in many places. In this university, how-

ever, granolithic has been used in the chemical laboratories for 15 years with excellent satisfaction. It should be noted, further, that in the mechanical and engineering laboratories the floor rests directly on the ground, while in the chemical laboratory there is a warm room or basement underneath.

12 The life of a well laid granolithic surface under foot traffic is practically permanent. Tests of various materials and methods are discussed elsewhere in this paper, and specifications are given in appendices for durable granolithic surfaces.

13 *Granolithic with Ground Surface.* Experimental surfaces, together with laboratory tests made as a check, show that a pleasing surface, approaching terrazzo in appearance and fully as durable under foot traffic, can be obtained by placing granolithic with scarcely any trowelling, and then grinding the surface just enough to expose the grains of sand and stone. The grains which show are finer than in terrazzo and darker colored. The appearance, however, is pleasing. Removal of the scum takes away the monotony of the plain gray cement surface, since this is relieved by the various colors of the sand and stone. A glossy effect can be produced if desired by the grinding which permits of easy cleaning and gives a surface suitable even for a lavatory at much less cost than tile or terrazzo. Still further to give variety to the appearance, tile can be placed in patterns or as a border.

14 The University of Missouri, which refers to the dust from granolithic floors, believes that this difficulty would be solved by grinding the surface instead of trowelling. Specifications giving the method of laying the concrete granolithic and grinding it are presented at the end of this paper. From observations of the time required for grinding the surfaces and allowing amply for delays, the extra cost for grinding is estimated not to exceed 3c. per square foot of surface area.

15 *Linoleum.* The hardness and noise characteristic of granolithic finish are overcome by covering the surface with Battleship linoleum. In the few colleges where this has been adopted they are very enthusiastic over the results. In other places, such as offices, the same type of construction meets with a great deal of favor. At the University of Chicago cork carpets are used, which answer a similar purpose.

16 Linoleum is laid on a concrete surface, which need not be brought to a fine finish and therefore can be completed at the time

the base concrete is laid and at a low cost. Any marring of the surface or sudden rains will not affect its use for the linoleum finish.

17 The linoleum should be stuck firmly to the granolithic surface and preferably a cove base should be run around the room and sills provided at entrances so that the surface of the granolithic will be flush. In this way the edges are prevented from fraying. The life of first-class quality Battleship linoleum, if edges are not frayed, is probably from 15 to 30 years, depending upon the amount of travel. These ages are estimated from records of linoleum now in use.

18 Linoleum, after allowing for the better finish required on the concrete, costs substantially the same as a single floor of birch or maple, but it is noiseless, more uniform in appearance, and requires less labor for maintenance in good condition. Its superiority to wood is indicated by the fact that wood floors are frequently covered with linoleum.

19 *Hardwood Floors.* Floors of maple, birch, beech, oak, or long leafed Southern pine are used most largely for offices, classrooms, or lecture rooms, and in many of the older colleges for laboratories and halls. A wood surface, however, is not usually considered entirely satisfactory either in general appearance or in wearing qualities. If one passes from a corridor with a granolithic, terrazzo, or tile floor, into a room or auditorium having a wood floor, there is a marked effect of inferiority and cheapness. There is just as much danger of poor materials and workmanship with wood as with other kinds of floors. Unless the greatest care is taken in selection of materials and workmanship, they are liable to shrink or swell and sometimes to squeak under foot. If at all hollow underneath, they are more noisy than a concrete surface. The floors of the New Grand Central office buildings are an example of this.

20 For corridors, wood is being largely superseded by granolithic, terrazzo, or tile. For laboratories other materials are being substituted for wood in most of the newer structures, although wood is occasionally preferred, especially for physical laboratories and for laboratories where men stand for long periods. The linoleum on concrete will overcome practically all the objections that are made to wood floors, with a cost substantially the same.

21 There are various methods of laying hardwood floors. For classrooms a single thickness of maple or birch nailed to sleepers with cinder concrete between should be satisfactory. Another type of construction is to use patented metal screeds embedded in the

base concrete, and nail the floor boards to splines in the screeds. For rooms subjected to heavy traffic, 2-in. or 2½-in. plank may be placed underneath the hardwood floor.

22 Of all the different materials, oak is the most expensive and the finest in appearance at the beginning, but under heavy traffic is more liable to splinter than the finer grained woods. Georgia pine, if of best quality, makes a durable floor, and is preferable to the finer grained woods in wet places, as it does not swell and warp so badly. It is less durable, however, and therefore not recommended for the greatest permanence in rooms such as class and lecture rooms. Maple, birch, and beech, all make good floor material. These are usually laid in strips ⅞-in. thick by 2¼-in. wide. The quality varies largely, ranging in cost from $32 per 1000 to $75 per 1000.

23 *Terrazzo.* Terrazzo is made by spreading upon the base concrete a mixture of neat cement and marble chips and grinding the surface to a depth sufficient to cut into stones and expose them on their largest diameters. Marble, sometimes white and sometimes colored, is used, and since no sand is employed the particles may be of fairly uniform size. The joints between the particles being of neat cement are hard and even more durable than the pieces of the marble themselves. Large pieces of marble, from ¾-in. to 1-in. in diameter, give a more distinctive floor but cost more than a floor of the smaller stones, from ⅛-in. to ½-in. in diameter, because the large stones require much more grinding to get down to the large diameters of the particles. There is more tendency to crack than in a good granolithic properly bonded to the base, but if laid with the best workmanship, this cracking is reduced to a minimum.

24 Terrazzo is largely used, especially in the newer office buildings and in institutions, for corridors and halls. It also is satisfactory for lavatories, although more expensive than granolithic. It appears from our investigation that for both of these uses concrete with a ground surface can be substituted at less cost and with satisfactory results.

25 In certain cases objection—which applies also to any hard material like granolithic or tile—is raised to terrazzo because of the noise, and even corridors are covered with linoleum or similar material.

26 *Marble Mosaic.* Mosaic consists of small squares of marble laid on the cement bed, something like terrazzo. Surfaces are

ground enough to make all pieces true and level. The price of mosaic is too high to be considered for large areas and in many cases the pieces of marble pull out from the surface. Mosaic is suitable in certain cases for an ornamental border which is not subject to wear.

27 *Magnesium Composition.* When laid with great care, composition is a satisfactory and durable material. Floors 6 or 8 years old have been examined and show satisfactory wear. The work must be done by a responsible firm with a suitable guarantee bond, because even with the greatest care the work is occasionally imperfect. The imperfections, however, are apt to show within the first year of service. Composition is more resilient than granolithic, so that there are less complaints of hardness. It is nearly, but not quite, as noisy. Furniture can be screwed directly to the composition.

28 Composition has not yet been used to a great extent in colleges. The floors of Cooper Union in New York City are many of them covered with this material and the results have been satisfactory. It is suitable for certain laboratories, such as physical and biological.

29 *Asphalt Composition.* Asphalt composition is suitable for certain places where no heavy tools or machines are liable to press into the soft surface. It is resilient and easy to walk and stand upon. The color is not pleasing, being a dead black. In a few colleges it has been used satisfactorily for chemical laboratories. At Harvard, for example, asphalt mastic on top of wood has been in satisfactory use for many years. Johns Hopkins considered this material for their new chemical laboratories but abandoned it because of its viscous properties, substituting granolithic finish, which has proved satisfactory.

30 *Tiles.* Tile of various colors is an excellent material for corridors, lavatories, and even for laboratories, but is too expensive to use except where required for architectural treatment. There are various types and qualities of tile, ranging from quarry tile to cork and rubber tile. All of them, however, are expensive.

31 *Wood Block.* Wood block may be suitable in certain cases for a basement floor having severe usage. In the University of Cincinnati wood block is used in the mechanical and electrical testing laboratories and appears to be satisfactory.

TESTS AND INVESTIGATIONS OF GRANOLITHIC FLOORS

32 The material used most largely for floor surfaces in factory construction and also to a considerable extent in other structures is what is termed a granolithic surface. This, as generally understood, is a layer of mortar or concrete from ½-in. to 2-in. thick, usually about 1 in. on top of the concrete slab and bonded to it. Although granolithic or concrete floors are so widely employed, neither the materials nor the methods of construction are standardized and scarcely two contractors or engineers adopt the same methods. Moreover, the materials available in a given locality largely affect the choice.

33 In order to compare the materials, that is, the aggregates, available for new Technology, and to determine the best proportions and methods of laying these materials, a series of sample surfaces were laid at the factory of the Simplex Wire & Cable Company, in Cambridge. Also, comparative tests were made with similar materials in other locations. A few preliminary laboratory tests were carried through, and certain tests to determine the best method of bonding a new granolithic surface to a hardened concrete base. As a result of these experiments, the following recommendations are made for the granolithic finish of floors for which this material is to be used. The conclusions apply also to structures in general.

34 *Materials.* The various aggregates used in the tests include three kinds of sand mixed as mortars in different proportions, and combinations of these sands with samples of different granites and traps. One or two sections were also laid with a patented compound.

35 Careful examination and comparisons of the various sections of slab with reference to hardness and appearance led to the selection of Plum Island sand, which should be specified to have not more than 10 per cent of its grains pass a sieve having 50 meshes to the linear inch, and not more than 2 per cent pass a sieve having 100 meshes to the linear inch; and crushed granite of a size which has passed the ¾-in. screen in a crusher plant and been caught on the $^3/_{16}$-in. screen.

36 As a result of this selection, a slab of considerable area was laid at a later date at the Simplex factory with the selected materials and proportions, and in a position where it would receive rather hard usage. The Simplex Company have recently advised us that they consider this slab the best piece of granolithic that has been laid in the factory.

37 *Proportions.* Different proportions of the materials were employed in the various sample sections, each of which was about 2 ft. wide by 3 ft. long. The principal proportions tested were 1:2 with sand alone; 1:1½ with sand alone; 1:1:1¼ with sand and fine crushed stone; 1:1:1½ with the same materials, and 1:¾:1¼. As a result the proportions selected as best are one part cement to ¾ parts Plum Island sand to 1¼ parts crushed granite.

38 *Method of Laying Granolithic.* Instead of using a soft, flowing mixture, the best results were obtained by using a fairly stiff mixture, stiff enough to be rammed in place by a square-faced rammer, which would bring the mortar readily to the surface. In this way the surface skin is thinner, there is less liability to dust, and the body of the concrete, which is of a better quality than with a wetter mix, is reached with comparatively little wear, so that the dusting does not continue.

39 *Treatment of Surfaces.* Dusting is temporarily overcome by paint, but this is always unsatisfactory because it wears off under ordinary travel, and if the concrete is not of the best quality it then begins to dust. With the adoption of the specifications in the Appendix no surface material should be needed.

40 *Grinding Surfaces of Granolithic.* Objections to granolithic finish are dusting of the surface, the dead gray color, and the liability of local defects. Experiments show that these can be overcome by grinding the surfaces with a carborundum machine. This method was followed on a section of slab at the Simplex Wire & Cable Company.

41 The general plan adopted is similar to that used with terrazzo finish. Instead, however, of grinding off a considerable thickness and thus entailing a large expense per square foot, only a very thin layer is taken off so as to show the grains of sand and the pieces of coarser aggregate.

42 With this treatment, the surface is of a varied texture, and shows the various colored grains, and permits of different effects by using aggregates of different colors. While the effect is not so conspicuous as the terrazzo, the surface is of a quieter tone, and should be satisfactory for ordinary corridors and halls. The grinding renders the surface more glossy and denser, so that it is possible to use this treatment with good results in a lavatory or other places where frequent washing and cleaning is required. To produce a more ornamental effect, borders or patterns of tile may be placed in the concrete.

43 *Bond of Granolithic to Base Concrete.* A perfect bond between the granolithic and the base concrete is obtained most easily by placing the granolithic before the base concrete has reached its set. Surfaces thus laid are liable to injury from the workmen who have to go upon them before they have hardened thoroughly, and occasionally an unexpected shower will roughen the surface in such a way that it is very difficult to repair. To determine the best method of bonding, one which would give thorough assurance of perfect adhesion, tests were made and then tried out in the field on a large concrete building.

44 Laboratory tests were made on bonding new mortar to an old concrete surface, using various methods of treatment of surface, ncluding acid treatment, roughening, and no surface treatment whatever. As bonding material, neat cement was used in different conditions of plasticity, also certain patented compounds. As a result of these tests and experience in the field, a roughened surface of the old concrete, with neat cement paste brushed in, is recommended as an effective method to produce a positive bond.

45 Specifications for bonding are given in the Appendix. It was shown in the tests that with a proper neat cement bond on a roughened surface the break under tension was frequently through the concrete rather than at the joint.

46 *Preparing Concrete Base for other Surface Materials.* If some other material than granolithic is used for the wearing surface, the base concrete must be left in a condition satisfactory for placing the surface. For most materials, such as hardwood finish, composition, asphalt, and similar treatments, the surface of the base must be brought more nearly level than where granolithic is used. This can be accomplished by very careful screeding of the surface, trowelling of rough places, and filling holes made by footprints before the concrete has hardened. An allowance of 1 cent per sq. ft. is made in the cost estimates for this extra treatment.

47 For linoleum, a real granolithic is not required, but the surface must be level and true. This should be accomplished by spreading a thin layer of mortar before the base concrete is set, but this need not be of the very best quality of granolithic unless with the object of using portions of the floor without linoleum. This thickness of the mortar may be ½ in. to ¾ in. This should be trowelled at the proper periods, but with less care than for a granolithic that is to be used as wearing surface. Some form of cove base around the walls is advantageous to use with the linoleum.

APPENDIX

SPECIFICATIONS FOR LAYING GRANOLITHIC FINISH ON SET CONCRETE

48 Specifications for laying granolithic finish on set concrete are as follows:

a Roughen surface of base concrete at the age of about 24 hours, so as to remove most of surface scum.

b If surfaces have not been thus roughened, pick with a bushhammer to remove a part but not all of the surface skin.

c Spread dilute muriatic acid about one part acid to four parts water over the surface, allow to stand for a few minutes, then soak thoroughly with water, and wash off the surface.

d Sweep off the excess water on the surface of the concrete and spread on a coating about ⅛ in. thick of neat cement paste, and broom it well into the concrete. (Do not use dry cement for this.)

e Mix the granolithic in proportions 1 part cement to ¾ parts coarse sand, like Plum Island, to 1¼ part crushed granite screened through a ¾-in. screen and caught on 3/16-in. dust jacket.

f Make the consistency of granolithic rather stiff so that the mortar will just flush to the surface.

g Have the screeds laid parallel and level so that the granolithic can be spread even with straight-edge. Run over the screeds. See that plenty of material is being pushed ahead of the straight-edge at all times so as to avoid pockets in the surface.

h Ram granolithic with light square-faced tamper.

i Float granolithic surface as soon as it begins to stiffen.

j Trowel granolithic surface hard as soon as the proper stage has been reached. (If surface is to be ground do not give surface this final trowelling.)

k Cover the surfaces of the granolithic about 24 hours after laying, with wet burlap or similar material which will hold water. Wet material each day, and oftener if necessary, for a period of 14 days.

GENERAL REQUIREMENT

49 Never lay concrete finish in cold weather unless a uniform temperature can be maintained by artificial heat, as the cold prevents the surface of the granolithic from hardening satisfactorily. In laying floors where water is to be used, care should be exercised to provide the required slope for cleaning and drainage. This is especially necessary in such places as chemical laboratories.

GRINDING GRANOLITHIC SURFACES

50 Specifications for granolithic grinding surfaces are as follows:

a Lay the granolithic as described in Appendix No. 1, but omit the final trowelling.

b Rub the granolithic surface by hand with carborundum block at the age of about 24 hours after placing. Rub lightly and take off only the top scum of the cement and remove any surface irregularities.

c Grind the surfaces with a floor polishing machine at the age of about 7 days (time varies with weather and temperature). Use about 60–80 grit with water and do not use any sand unless it is found necessary. This grinding should take off the top film of the surface and cut into the sand grains enough to expose them and to leave the surface smooth but not shiny.

d Rub wet cement paste into any pinholes.

DISCUSSION

In presenting the paper THE AUTHOR stated that good granolithic floors were being built which would stand very severe traffic and dust only to a very small degree. The five principal requirements are, materials, proportions, bonding, methods of laying, and treatment of surface. Use coarse material with the cement, avoiding fine sand or stone with fine particles because this rises to the surface in trowelling. Use a comparatively dry mix that will require a slight tamping. There are certain compounds on the market that will prevent dusting, but if the right materials and workmanship are employed, so little dusting will occur that it will not be objectionable for ordinary uses.

For durability it is best to lay the floor surface along with the base concrete. This is often impracticable and tests have proven that a good bond can be obtained on old concrete with the proper treatment. Special attention is called to the specifications given at the close of the paper and to the note with reference to laying granolithic in cold weather.

Ross F. TUCKER[1] (written). So much difficulty exists in securing a good wearing surface for granolithic that hardwood is preferred for all purposes, particularly where operatives have to be on their feet.

L. C. WASON[2] (written). In regard to costs, my figures for granolithic construction would in each case be 2 cents less than those quoted by the author. For hardwood floors, there is a difference of 3¾ cents per sq. ft. between 2 in. face by 1⅛ in. thick and 3½ in. face by ⅞ in. thick, in both cases the material costing

[1]Consulting Engineer, 35 W. 32d St., New York.

[2]Pres. and Engr., Aberthaw Constr. Co., 8 Beacon St., Boston, Mass.

the same per thousand. Magnesium composition is also laid at a third less than quoted.

Lameness and fatigue, which factory operatives thought were caused by the hardness of granolithic floors, are due to the fact that such floors are better conductors of heat. This has been overcome where floors are cold by wearing heavy shoes.

Nalecod, another material composed of asbestos, portland cement and sand, is giving better results than screed for wood floors.

For a granolithic floor no particles should be used smaller than those passing a No. 30 sieve, and hard rock which will withstand abrasion should be used, without any sand, the proportion being one to two. In bonding to old surfaces, a thin top i.e., ¾ in. or 1 in., is more likely to come loose than a thicker one.

Commenting on paragraph 48, a multiple pick is cheaper and requires a less experienced workman than a bush hammer.

Dilute muriatic acid is unsafe unless the concrete is dense, otherwise it is likely to soften it.

The mixture should not be limited to granite, as traps and gravel give good results. A stiff mortar gives best results, although a different consistency should be used, depending on whether the base is fully set.

G. S. Walker. There is almost certain to be trouble with granolithic if the base is allowed to set and an attempt afterward made to bond the surface to it. This is due to the shrinkage rate being different. They should always be laid together.

Walter S. Timmis. There is a very serious defect in wood floors in fireproof buildings, that of springiness. This was more apparent in the earlier buildings where floors were laid directly on the arches, but even in recent buildings it occurs, owing to lack of care in bringing the cinder fill to the top of the screed. Dry rotting of screed also often takes place, especially when the floor is laid before the cement is dry. On granolithic floors, the difficulty in getting a smooth surface and no dusting is due to the trowelling not being done at the psychological moment.

G. P. Hemstreet (written). I desire to call attention to another form of asphalt floor for fireproof buildings, streets, piers, warehouses, etc., used for many years, viz., asphalt blocks composed of hard crushed stone and about 7 per cent asphaltic cement formed under hydraulic pressure of various sizes and forms. These are laid

in cement mortar with joints grouted. The surface is smooth, resilient, non-slipping, dustless, sanitary and not easily marred. This material can be taken up whenever desired and readily replaced. The cost is $1.50 to $2.50 per square yard.

THE AUTHOR, in closing, recognizes the force of Mr. Wason's requirement that the aggregates should be practically free from grains finer than a No. 30 sieve. Unless a sand consisting of very coarse grains, like Plum Island sand, is available, he agrees that it is proper to use an aggregate with no sand but with particles graded from ¾ inch down to, say, a No. 30 sieve.

Regarding Mr. Walker's criticism on bonding new granolithic surfaces to old concrete, the best answer is that it has been done with good results by the methods described.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

REPORT

OF THE COMMITTEE TO FORMULATE STANDARD SPECIFICATIONS FOR THE CONSTRUCTION OF STEAM BOILERS AND OTHER PRESSURE VESSELS AND FOR THEIR CARE IN SERVICE

KNOWN AS

THE BOILER CODE COMMITTEE

RULES FOR THE CONSTRUCTION OF STATIONARY BOILERS AND FOR ALLOWABLE WORKING PRESSURES

Edition of 1914 with Index

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

To the Council of The American Society of Mechanical Engineers

Gentlemen: Your Committee appointed September 15th, 1911 to "Formulate Standard Specifications for the Construction of Steam Boilers and Other Pressure Vessels and for Care of Same in Service" respectfully submits its final report on Rules for the construction and allowable working pressures of stationary boilers, which forms a portion of the task assigned to it.

The primary object of these Rules is to secure safe boilers. The interests of boiler users and manufacturers have been carefully considered and the requirements made such that they will not entail undue hardship by departing too widely from present practice.

Your Committee recommends that you appoint a permanent committee to make such revisions as may be found desirable in these Rules, and to modify them as the state of the art advances, and that such committee should hold meetings at least once in two years at which all interested parties may be heard.

Yours truly,

John A. Stevens, *Chairman*
Wm. H. Boehm
Rolla C. Carpenter
Richard Hammond
Chas. L. Huston
Edward F. Miller
H. C. Meinholtz*
E. D. Meier*

COMMITTEE

Deceased*

Submitted to the Council of The American Society of Mechanical Engineers, February 13, 1915.

Your Committee secured the assistance of the following Engineers as an Advisory Committee, representing various phases of the design, installation and operation of boilers and the Rules were unanimously approved by them.

F. H. Clark, Railroad Sub-Committee, The American Society of Mechanical Engineers.

F. W. Dean, Consulting Engineers.

Thos. E. Durban, Boiler Manufacturers' Association, Uniform Specifications Committee, for all types of boilers.

Carl Ferrari, National Tubular Boiler Manufacturers' Association.

Elbert C. Fisher, Scotch marine and other types of boilers.

Arthur M. Greene, Jr., Engineering Education.

Chas. E. Gorton, Steel heating boilers.

A. L. Humphrey, Railroad Sub-Committee, The American Society of Mechanical Engineers.

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RULES FOR THE CONSTRUCTION OF STATIONARY BOILERS AND FOR ALLOWABLE WORKING PRESSURES

These Rules do not apply to boilers which are subject to federal inspection and control, including marine boilers, boilers of steam locomotive and other self-propelled railroad apparatus.

The Rules are divided into two parts:

PART I applies to new installations. { Section I, Power Boilers. Section II, Heating Boilers.

PART II applies to existing installations.

PART I NEW INSTALLATIONS

SECTION I

POWER BOILERS

Selection of Materials

1 Specifications are given in these Rules for the important materials used in the construction of boilers, and where given, the materials shall conform thereto.

2 Steel plates for any part of a boiler when exposed to the fire or products of combustion, and under pressure, shall be of firebox quality as designated in the Specifications for Boiler Plate Steel.

3 Steel plates for any part of a boiler, where firebox quality is not specified, when under pressure, shall be of firebox or flange quality as designated in the Specifications for Boiler Plate Steel.

4 Braces when welded, shall be of wrought-iron of the quality designated in the Specifications for Refined Wrought-Iron Bars.

5 Manhole and handhole covers and other parts subjected to pressure and braces and lugs, when made of steel plate, shall be of firebox or flange quality as designated in the Specifications for Boiler Plate Steel.

6 Steel bars for braces and for other boiler parts, except as otherwise specified herein, shall be of the quality designated in the Specifications for Steel Bars.

7 Staybolts shall be of iron or steel of the quality designated in the Specifications for Staybolt Iron or in the Specifications for Staybolt Steel.

8 Rivets shall be of steel or iron of the quality designated in the Specifications for Boiler Rivet Steel or in the Specifications for Boiler Rivet Iron.

9 Cross pipes connecting the steam and water drums of water-tube boilers, headers and cross boxes and all pressure parts of the boiler proper over 2-in. pipe size, or equivalent cross-sectional area, shall be of wrought steel, or cast steel of Class B grade, as designated in the Specifications for Steel Castings, when the maximum allowable working pressure exceeds 160 lb. per sq. in.

10 Mud drums of boilers used for other than heating purposes shall be of wrought steel, or cast steel of Class B grade, as designated in the Specifications for Steel Castings.

11 Pressure parts of superheaters, separately fired or attached to stationary boilers, unless of the locomotive type, shall be of wrought steel, or cast steel of Class B grade, as designated in the Specifications for Steel Castings.

12 Cast iron shall not be used for boiler and superheater mountings, such as nozzles, connecting pipes, fittings, valves and their bonnets, for steam temperatures of over 450 deg. fahr.

13 Water-leg and door-frame rings of vertical fire-tube boilers 36 in. or over in diameter, and of locomotive and other type boilers, shall be of wrought iron or steel, or cast steel of Class B grade, as designated in the Specifications for Steel Castings. The O G or other flanged construction may be used as a substitute in any case.

Ultimate Strength of Material used in Computing Joints

14 *Tensile Strength of Steel Plate.* The tensile strength used in the computations for steel plates shall be that stamped on the plates as herein provided, which is the minimum of the stipulated range, or 55,000 lbs. per sq. in. for all steel plates, except for special grades having a lower tensile strength.

15 *Crushing Strength of Steel Plate.* The resistance to crushing of steel plate shall be taken at 95,000 lb. per sq. in. of cross-sectional area.

16 *Strength of Rivets in Shear.* In computing the ultimate

strength of rivets in shear, the following values in pounds per square inch of the cross-sectional area of the rivet shank shall be used:

Iron rivets in single shear	38,000
Iron rivets in double shear	76,000
Steel rivets in single shear	44,000
Steel rivets in double shear	88,000

The cross-sectional area used in the computations shall be that of the rivet shank after driving.

Minimum Thicknesses of Plates and Tubes

17 *Thickness of Plates.* The minimum thickness of any boiler plate under pressure shall be ¼ in.

18 The minimum thicknesses of shell plates, and dome plates after flanging, shall be as follows:

When the Diameter of Shell is			
36 In. or Under	Over 36 In. to 54 In.	Over 54 In. to 72 In.	Over 72 In.
¼ in.	5/16 in.	⅜ in.	½ in.

19 The minimum thicknesses of butt straps shall be as given in Table 1.

TABLE 1 MINIMUM THICKNESSES OF BUTT STRAPS

Thickness of Shell Plates, In.	Minimum Thickness of Butt Straps, In.	Thickness of Shell Plates, In.	Minimum Thickness of Butt Straps, In.
¼	¼	11/16	7/16
9/32	¼	¾	7/16
5/16	¼	13/16	½
11/32	¼	⅞	½
⅜	5/16	⅞	5/8
13/32	5/16	1	¾
7/16	⅜	1⅛	¾
15/32	⅜	1¼	⅞
½	7/16		

20 The minimum thicknesses of tube sheets for horizontal return tubular boilers, shall be as follows:

When the Diameter of Tube Sheet is			
42 In. or Under	Over 42 In. to 54 In.	Over 54 In. to 72 In.	Over 72 In.
⅜ in.	7/16 in.	½ in.	9/16 in.

21 *Tubes for Water-Tube Boilers.* The minimum thicknesses of tubes used in water-tube boilers measured by Birmingham wire gage, for maximum allowable working pressures not exceeding 165 lb. per sq. in., shall be as follows:

Diameters less than 3 in	No. 12 B.W.G.
Diameter 3 in. or over, but less than 4 in	No. 11 B.W.G.
Diameter 4 in. or over, but less than 5 in	No. 10 B.W.G.
Diameter 5 in	No. 9 B.W.G.

The above thicknesses shall be increased for maximum allowable working pressures higher than 165 lb. per sq. in. as follows:

Over 165 lb. but not exceeding 235 lb	1 gage
Over 235 lb. but not exceeding 285 lb	2 gages
Over 285 lb. but not exceeding 400 lb	3 gages

Tubes over 4-in. diameter shall not be used for maximum allowable working pressures above 285 lb. per sq. in.

22 *Tubes for Fire-Tube Boilers.* The minimum thicknesses of tubes used in fire tube boilers measured by Birmingham wire gage, for maximum allowable working pressures not exceeding 175 lb. per sq. in., shall be as follows:

Diameters less than 2½ in	No. 13 B.W.G.
Diameter 2½ in. or over, but less than 3¼ in	No. 12 B.W.G.
Diameter 3¼ in. or over, but less than 4 in	No. 11 B.W.G.
Diameter 4 in. or over, but less than 5 in	No. 10 B.W.G.
Diameter 5 in	No. 9 B.W.G.

For higher maximum allowable working pressures than given above the thicknesses shall be increased one gage.

SPECIFICATIONS FOR BOILER PLATE STEEL

THESE SPECIFICATIONS[1] ARE SIMILAR TO THOSE OF THE AMERICAN SOCIETY FOR TESTING MATERIALS, SERIAL DESIGNATION A **30-14.**

23 *Grades.* These specifications cover two grades of steel for boilers, namely: FLANGE and FIREBOX.

I Manufacture

24 *Process.* The steel shall be made by the open-hearth process.

II Chemical Properties and Tests

25 *Chemical Composition.* The steel shall conform to the following requirements as to chemical composition:

	FLANGE	FIREBOX
Carbon	.....	Plates ¾ in. thick and under. . 0.12—0.25 per cent
		Plates over ¾ in. thick. . . . . 0.12—0.30 per cent
Manganese	0.30—0.60 per cent	0.30—0.50 per cent
Phosphorus, Acid	Not over 0.05 per cent	Not over 0.04 per cent
Phosphorus, Basic	Not over 0.04 per cent	Not over 0.035 per cent
Sulphur	Not over 0.05 per cent	Not over 0.04 per cent
Copper	.....	Not over 0.05 per cent

26 *Ladle Analyses.* An analysis shall be made by the manufacturer from a test ingot taken during the pouring of each melt, a copy of which shall be given to the purchaser or his representative. This analysis shall conform to the requirements specified in Par. 25.

27 *Check Analyses.* Analyses may be made by the purchaser from a broken tension test specimen representing each plate as rolled, which shall conform to the requirements specified in Par. 25.

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[1]Approved and recommended in its modified form, October 9, 1914, by the Association of American Steel Manufacturers, the American Boiler Manufacturers' Association, the National Tubular Boiler Manufacturers' Association, the National Association of Thresher Manufacturers and the representatives present of leading Water Tube Boiler Manufacturers, with whom the Boiler Code Committee was in conference on September 16, 1914, and by whom further modifications were afterwards offered.

III Physical Properties and Tests

28 *Tension Tests.* *a* The material shall conform to the following requirements as to tensile properties:

	FLANGE	FIREBOX
Tensile strength, lb. per sq. in.	55,000—65,000	55,000—63,000
Yield point, min., lb. per sq. in.	0.5 tens. str.	0.5 tens. str.
Elongation in 8-in., min., per cent (See Par. 29)	1,500,000 / Tens. str.	1,500,000 / Tens. str.

b If desired steel of lower tensile strength than the above may be used in an entire boiler, or part thereof, the desired tensile limits to be specified, having a range of 10,000 lb. per sq. in. for flange or 8000 lb. per sq. in. for firebox, the steel to conform in all respects to the other corresponding requirements herein specified, and to be stamped with the minimum tensile strength of the stipulated range.

c The yield point shall be determined by the drop of the beam of the testing machine.

29 *Modifications in Elongation.* *a* For material over ¾ in. in thickness, a deduction of 0.5 from the percentages of elongation specified in Par. 28*a*, shall be made for each increase of ⅛ in. in thickness above ¾ in., to a minimum of 20 per cent.

b For material ¼ in. or under in thickness, the elongation shall be measured on a gage length of 24 times the thickness of the specimen.

30 *Bend Tests.* *a* *Cold-bend Tests*—The test specimen shall bend cold through 180 deg. without cracking on the outside of the bent portion, as follows: For material 1 in. or under in thickness, flat on itself; and for material over 1 in. in thickness, around a pin the diameter of which is equal to the thickness of the specimen.

b *Quench-bend Tests*—The test specimen, when heated to a light cherry red as seen in the dark (not less than 1200 deg. fahr.), and quenched at once in water the temperature of which is between 80 deg. and 90 deg. fahr., shall bend through 180 deg. without cracking on the outside of the bent portion, as follows: For material 1 in. or under in thickness, flat on itself; and for material over 1 in. in thickness, around a pin the diameter of which is equal to the thickness of the specimen.

31 *Homogeneity Tests.* For firebox steel, a sample taken from a broken tension test specimen shall not show any single seam or cavity more than ¼ in. long, in either of the three fractures obtained in the test for homogeneity, which shall be made as follows:

The specimen shall be either nicked with a chisel or grooved on a machine, transversely, about 1/16 in. deep, in three places about 2 in. apart. The first groove shall be made 2 in. from the square end; each succeeding groove shall be made on the opposite side from the preceding one. The specimen shall then be firmly held in a vise, with the first groove about ¼ in. above the jaws, and the projecting end broken off by light blows of a hammer, the bending being away from the groove. The specimen shall be broken at the other two grooves in the same manner. The object of this test is to open and render visible to the eye any seams due to failure to weld or to interposed foreign matter, or any cavities due to gas bubbles in the ingot. One side of each fracture shall be examined and the length of the seams and cavities determined, a pocket lens being used if necessary.

32 *Test Specimens.* Tension and bend test specimens shall be taken from the finished rolled material. They shall be of the full

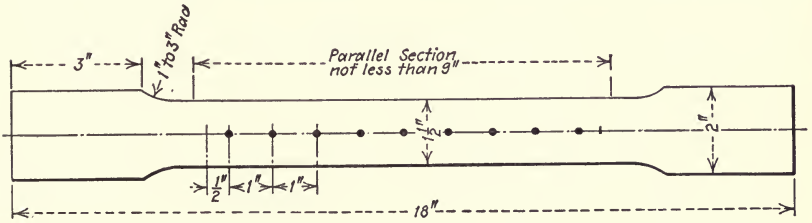

FIG. 1 STANDARD FORM OF TEST SPECIMEN REQUIRED FOR ALL TENSION TESTS OF PLATE MATERIAL

thickness of material as rolled, and shall be machined to the form and dimensions shown in Fig. 1; except that bend test specimens may be machined with both edges parallel.

33 *Number of Tests.* *a* One tension, one cold-bend, and one quench-bend test shall be made from each plate as rolled.

b If any test specimen shows defective machining or develops flaws, it may be discarded and another specimen substituted.

c If the percentage of elongation of any tension test specimen is less than that specified in Pars. 28 and 29, and any part of the fracture is outside the middle third of the gaged length, as indicated by the scribe scratches marked on the specimen before testing, a retest shall be allowed.

IV PERMISSIBLE VARIATION IN GAGE

34 *Permissible Variation.* The thickness of each plate shall not vary under the gage specified more than 0.01 in. The overweight

limits are considered a matter of contract between the steel manufacturer and the boiler builder.

V Finish

35 *Finish.* The finished material shall be free from injurious defects and shall have a workmanlike finish.

VI Marking

36 *Marking.* *a* Each shell plate shall be legibly stamped by the manufacturer with the melt or slab number, name of manufacturer, grade and the minimum tensile strength of the stipulated range as specified in Par. 28, in three places, two of which shall be located at diagonal corners about 12 in. from the edge and one about the center of the plate, or at a point selected and designated by the purchaser so that the stamp shall be plainly visible when the boiler is completed.

b Each head shall be legibly stamped by the manufacturer in two places, about 12 in. from the edge, with the melt or slab number, name of manufacturer, grade, and the minimum tensile strength of the stipulated range as specified in Par. 28, in such manner that the stamp is plainly visible when the boiler is completed.

c Each butt strap shall be legibly stamped by the manufacturer in two places on the center line about 12 in. from the ends with the melt or slab number, name of manufacturer, grade, and the minimum tensile strength of the stipulated range as specified in Par. 28.

d The melt or slab number shall be legibly stamped on each test specimen.

VII Inspection and Rejection

37 *Inspection.* The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the material ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the material is being furnished in accordance with these specifications. All tests (except check analyses) and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

38 *Rejection.* *a* Unless otherwise specified, any rejection based on tests made in accordance with Par. 27 shall be reported within five working days from the receipt of samples.

b Material which shows injurious defects subsequent to its acceptance at the manufacturer's works will be rejected, and the manufacturer shall be notified.

39 *Rehearing.* Samples tested in accordance with Par. 27, which represent rejected material, shall be preserved for two weeks from the date of the test report. In case of dissatisfaction with the results of the tests, the manufacturer may make claim for a rehearing within that time.

SPECIFICATIONS FOR BOILER RIVET STEEL

THESE SPECIFICATIONS ARE SUBSTANTIALLY THE SAME AS THOSE OF THE AMERICAN SOCIETY FOR TESTING MATERIALS, SERIAL DESIGNATION A 31-14.

A Requirements for Rolled Bars

I Manufacture

40 *Process.* The steel shall be made by the open-hearth process.

II Chemical Properties and Tests

41 *Chemical Composition.* The steel shall conform to the following requirements as to chemical composition:

Manganese	0.30–0.50 per cent
Phosphorus	not over 0.04 per cent
Sulphur	not over 0.045 per cent

42 *Ladle Analyses.* An analysis to determine the percentages of carbon, manganese, phosphorus and sulphur shall be made by the manufacturer from a test ingot taken during the pouring of each melt, a copy of which shall be given to the purchaser or his representative. This analysis shall conform to the requirements specified in Par. 41.

43 *Check Analyses.* Analyses may be made by the purchaser from finished bars, representing each melt, which shall conform to the requirements specified in Par. 41.

III Physical Properties and Tests

44 *Tension Tests.* *a* The bars shall conform to the following requirements as to tensile properties:

Tensile strength, lb. per sq. in.	45,000–55,000
Yield point, min., lb. per sq. in.	0.5 tens. str.
Elongation in 8 in., min., per cent.	1,500,000
but need not exceed 30 per cent.	Tens. str.

b The yield point shall be determined by the drop of the beam of the testing machine.

45 *Bend Tests.* *a* *Cold-bend Tests*—The test specimen shall bend cold through 180 deg. flat on itself without cracking on the outside of the bent portion.

b *Quench-bend Tests*—The test specimen, when heated to a light cherry red as seen in the dark (not less than 1200 deg. fahr.), and quenched at once in water the temperature of which is between 80 deg. and 90 deg. fahr., shall bend through 180 deg. flat on itself without cracking on the outside of the bent portion.

46 *Test Specimens.* Tension and bend test specimens shall be of the full-size section of bars as rolled.

47 *Number of Tests.* *a* Two tension, two cold-bend, and two quench-bend tests shall be made from each melt, each of which shall conform to the requirements specified.

b If any test specimen develops flaws, it may be discarded and another specimen substituted.

c If the percentage of elongation of any tension test specimen is less than that specified in Par. 44 and any part of the fracture is outside the middle third of the gaged length, as indicated by scribe scratches marked on the specimen before testing, a retest shall be allowed.

48 *Permissible Variations in Gage.* The gage of each bar shall not vary more than 0.01 in. from that specified.

V Workmanship and Finish

49 *Workmanship.* The finished bars shall be circular within 0.01 in.

50 *Finish.* The finished bars shall be free from injurious defects and shall have a workmanlike finish.

VI Marking

51 *Marking.* Rivet bars shall, when loaded for shipment, be properly separated and marked with the name or brand of the manufacturer and the melt number for identification. The melt number shall be legibly marked on each test specimen.

VII Inspection and Rejection

52 *Inspection.* The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the bars ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the bars are being furnished in accordance with these specifications. All tests (except check analyses) and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

53 *Rejection.* *a* Unless otherwise specified, any rejection based on tests made in accordance with Par. 43 shall be reported within five working days from the receipt of samples.

b Bars which show injurious defects subsequent to their acceptance at the manufacturer's works will be rejected, and the manufacturer shall be notified.

54 *Rehearing.* Samples tested in accordance with Par. 43, which represent rejected bars, shall be preserved for two weeks from the date of the test report. In case of dissatisfaction with the results of the tests, the manufacturer may make claim for a rehearing within that time.

B Requirements for Rivets

I Physical Properties and Tests

55 *Tension Tests.* The rivets, when tested, shall conform to the requirements as to tensile properties specified in Par. 44, except that the elongation shall be measured on a gaged length not less than four times the diameter of the rivet.

56 *Bend Tests.* The rivet shank shall bend cold through 180 deg. flat on itself, as shown in Fig. 2, without cracking on the outside of the bent portion.

57 *Flattening Tests.* The rivet head shall flatten, while hot, to a diameter 2½ times the diameter of the shank, as shown in Fig. 3, without cracking at the edges.

58 *Number of Tests.* *a* When specified, one tension test shall be made from each size in each lot of rivets offered for inspection.

b Three bend and three flattening tests shall be made from each size in each lot of rivets offered for inspection, each of which shall conform to the requirements specified.

II Workmanship and Finish

59 *Workmanship.* The rivets shall be true to form, concentric, and shall be made in a workmanlike manner.

60 *Finish.* The finished rivets shall be free from injurious defects.

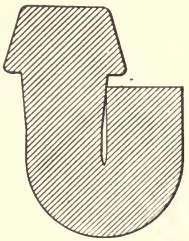

Fig. 2 The Bend Test for Rivets

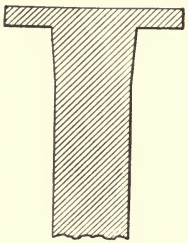

Fig. 3 The Flattening Test for Rivets

III Inspection and Rejection

61 *Inspection.* The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the rivets ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the rivets are being furnished in accordance with these specifications. All tests and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

62 *Rejection.* Rivets which show injurious defects subsequent to their acceptance at the manufacturer's works will be rejected, and the manufacturer shall be notified.

SPECIFICATIONS FOR STAYBOLT STEEL

REQUIREMENTS FOR ROLLED BARS

63 Steel for staybolts shall conform to the requirements for Boiler Rivet Steel specified in Pars. 40 to 62, except that the tensile properties shall be as follows:

Tensile strength, lb. per sq. in.	50,000–60,000
Yield point, min., lb. per sq. in.	0.5 tens. str.
Elongation in 8 in., min., per cent.	$\frac{1{,}500{,}000}{\text{Tens. str.}}$

Also with the exception that the permissible variations in gage shall be as follows:

Permissible Variations in Gage. The bars shall be truly round within 0.01 in. and shall not vary more than 0.005 in. above, or more than 0.01 in. below the specified size.

SPECIFICATIONS FOR STEEL BARS

THESE SPECIFICATIONS ARE ABSTRACTED FROM THOSE FOR STEEL FOR BRIDGES OF THE AMERICAN SOCIETY FOR TESTING MATERIALS, SERIAL DESIGNATION A 7-14.

I Manufacture

64 *Process.* The steel shall be made by the open-hearth process.

II Chemical Properties and Tests

65 *Chemical Composition.* The steel shall conform to the following requirements as to chemical composition:

Phosphorus	Acid	not over 0.06 per cent
	Basic	not over 0.04 per cent
Sulphur		not over 0.05 per cent

66 *Ladle Analysis.* An analysis to determine the percentages of carbon, manganese, phosphorus and sulphur shall be made by the manufacturer from a test ingot taken during the pouring of each melt, a copy of which shall be given to the purchaser or his representative. This analysis shall conform to the requirements specified in Par. 65.

III Physical Properties and Tests

67 *Tension Tests.* *a* The material shall conform to the following requirements as to tensile properties:

Tensile strength, lb. per sq. in.	55,000–65,000
Yield point, min., per sq. in.	0.5 tens. str.
Elongation in 8 in., min., per cent*	$\frac{1{,}500{,}000}{\text{Tens. str.}}$
Elongation in 2 in., min., per cent.	22

*See Par. 68.

b The yield point shall be determined by the drop of the beam of the testing machine.

68 *Modifications in Elongation.* *a* For bars over ¾ in. in thickness or diameter a deduction of 1 from the percentage of elongation in 8 in. specified in Par. 67, shall be made for each increase of ⅛ in. in thickness or diameter above ¾ in., to a minimum of 18 per cent.

b For bars under 5/16 in. in thickness or diameter a deduction of 2.5 from the percentage of elongation in 8 in. specified in Par. 67, shall be made for each decrease of 1/16 in. in thickness or diameter below 5/16 in.

69 *Bend Tests.* *a* The test specimen shall bend cold through 180 deg. without cracking on the outside of the bent portion, as follows: For material ¾ in. or under in thickness or diameter flat on itself; for material over ¾ in. to and including 1¼ in. in thickness or diameter around a pin the diameter of which is equal to the thickness or diameter of the specimen; and for material over 1¼ in. in thickness or diameter around a pin the diameter of which is equal to twice the thickness or diameter of the specimen.

b The test specimen for bars over 1½ in. in thickness or diameter when prepared as specified in Par. 70, shall bend cold through 180 deg. around a 1-in. pin without cracking on the outside of the bent portion.

70 *Test Specimens.* *a* Tension and bend test specimens except as specified in *b*, shall be of the full thickness of material as rolled. They may be machined to the form and dimensions shown in Fig. 1, or may have both edges parallel.

b Tension test specimens for bars over 1½ in. in thickness or diameter may be of the form and dimensions shown in Fig. 4. Bend

test specimens may be 1 by ½ in. in section. The axis of the specimen shall be located at any point midway between the center and surface and shall be parallel to the axis of the bar.

71 *Number of Tests.* *a* One tension and one bend test shall be made from each melt; except that if material from one melt differs ⅜ in. or more in thickness, one tension and one bend test shall be made from both the thickest and the thinnest material rolled.

b If any test specimen shows defective machining or develops flaws, it may be discarded and another specimen substituted.

c If the percentage of elongation of any tension test specimen is less than that specified in Par. 67, and any part of the fracture is more than ¾ in. from the center of the gage length of a 2-in. specimen or is outside the middle third of the gage length of an 8-in. specimen, as indicated by scribe scratches marked on the specimen before testing, a retest shall be allowed.

IV Permissible Variations in Gage

72 *Permissible Variation.* The thickness or cross-section of each piece of steel shall not vary under that specified more than 2.5 per cent. (Note: Overweight variation is a matter of contract between the steel manufacturer and boiler builder.)

V Finish

73 *Finish.* The finished material shall be free from injurious defects and shall have a workmanlike finish.

VI Marking

74 *Marking.* Bars shall, when loaded for shipment, be properly separated and marked with the name or brand of the manufacturer and melt number for identification. The melt number shall be legibly marked on each test specimen.

VII Inspection and Rejection

75 *Inspection.* The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works

which concern the manufacture of the material ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the material is being furnished in accordance with these specifications. All tests and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

76 *Rejection.* Material which shows injurious defects subsequent to its acceptance at the manufacturer's works will be rejected, and the manufacturer shall be notified.

SPECIFICATIONS FOR STEEL CASTINGS

THESE SPECIFICATIONS ARE ABSTRACTED FROM THOSE FOR STEEL CASTINGS OF THE AMERICAN SOCIETY FOR TESTING MATERIALS, SERIAL DESIGNATION A 27-14.

77 *Classes.* These specifications cover two classes of castings, namely:

Class A, ordinary castings for which no physical requirements are specified.

Class B, castings for which physical requirements are specified. These are of three grades: hard, medium, and soft.

78 *Patterns.* *a* Patterns shall be made so that sufficient finish is allowed to provide for all variations in shrinkage.

b Patterns shall be painted three colors to represent metal, cores, and finished surfaces. It is recommended that core prints shall be painted black and finished surfaces red.

79 *Basis of Purchase.* The purchaser shall indicate his intention to substitute the test to destruction specified in Par. 87, for the tension and bend tests, and shall designate the patterns from which castings for this test shall be made.

I Manufacture

80 *Process.* The steel may be made by the open-hearth, crucible, or any other process approved by the purchaser.

81 *Heat Treatment.* *a* Class *A* castings need not be annealed unless so specified.

b Class *B* castings shall be allowed to become cold. They shall then be uniformly reheated to the proper temperature to refine the

grain (a group thus reheated being known as an "annealing charge"), and allowed to cool uniformly and slowly. If, in the opinion of the purchaser or his representative, a casting is not properly annealed, he may at his option require the casting to be re-annealed.

II Chemical Properties and Tests

82 *Chemical Composition.* The castings shall conform to the following requirements as to chemical composition:

	Class A	Class B
Carbon	not over 0.30 per cent	.....
Phosphorus	not over 0.06 per cent	not over 0.05 per cent
Sulphur	.....	not over 0.05 per cent

83 *Ladle Analyses.* An analysis to determine the percentages of carbon, manganese, phosphorus and sulphur shall be made by the manufacturer from a test ingot taken during the pouring of each melt, a copy of which shall be given to the purchaser or his representative. This analysis shall conform to the requirements specified in Par. 82. Drillings for analysis shall be taken not less than ¼ in. beneath the surface of the test ingot.

84 *Check Analyses.* *a* Analyses of Class *A* castings may be made by the purchaser, in which case an excess of 20 per cent above the requirement as to phosphorus specified in Par. 82, shall be allowed. Drillings for analysis shall be taken not less than ¼ in beneath the surface.

b Analyses of Class *B* castings may be made by the purchaser from a broken tension or bend test specimen, in which case an excess of 20 per cent above the requirements as to phosphorus and sulphur specified in Par. 82, shall be allowed. Drillings for analysis shall be taken not less than ¼ in. beneath the surface.

III Physical Properties and Tests

(For Class *B* Castings only.)

85 *Tension Tests.* *a* The castings shall conform to the following minimum requirements as to tensile properties:

	Hard	Medium	Soft
Tensile strength, lb. per sq. in.	80,000	70,000	60,000
Yield point, lb. per sq. in.	36,000	31,500	27,000
Elongation in 2 in., per cent.	15	18	22
Reduction of area, per cent.	20	25	30

b The yield point shall be determined by the drop of the beam of the testing machine.

86 *Bend Tests.* *a* The test specimen for soft castings shall bend cold through 120 deg., and for medium castings through 90 deg., around a 1-in. pin, without cracking on the outside of the bent portion.

b Hard castings shall not be subject to bend test requirements.

87 *Alternative Tests to Destruction.* In the case of small or unimportant castings, a test to destruction on three castings from a lot may be substituted for the tension and bend tests. This test shall show the material to be ductile, free from injurious defects, and suitable for the purpose intended. A lot shall consist of all castings from one melt, in the same annealing charge.

88 *Test Specimens.* *a* Sufficient test bars, from which the test specimens required in Par. 89, may be selected, shall be attached to castings weighing 500 lb. or over, when the design of the castings will permit. If the castings weigh less than 500 lb., or are of such a design that test bars cannot be attached, two test bars shall be cast to represent each melt; or the quality of the castings shall be determined by tests to destruction as specified in Par. 87. All test bars shall be annealed with the castings they represent.

b The manufacturer and purchaser shall agree whether test bars can be attached to castings, on the location of the bars on the castings, on the castings to which bars are to be attached, and on the method of casting unattached bars.

c Tension test specimens shall be of the form and dimensions shown in Fig. 4. Bend test specimens shall be machined to 1 by ½ in. in section with corners rounded to a radius not over 1/16 in.

89 *Number of Tests.* *a* One tension and one bend test shall be made from each annealing charge. If more than one melt is represented in an annealing charge, one tension and one bend test shall be made from each melt.

b If any test specimen shows defective machining or develops flaws, it may be discarded; in which case the manufacturer and the purchaser or his representative shall agree upon the selection of another specimen in its stead.

c If the percentage of elongation of any tension test specimen is less than that specified in Par. 85, and any part of the fracture is more than ¾ in. from the center of the gaged length, as indicated by scribe scratches marked on the specimen before testing, a retest shall be allowed.

IV Workmanship and Finish

90 *Workmanship.* The castings shall substantially conform to the sizes and shapes of the patterns, and shall be made in a workmanlike manner.

91 *Finish.* *a* The castings shall be free from injurious defects.

b Minor defects which do not impair the strength of the castings may, with the approval of the purchaser or his representative, be

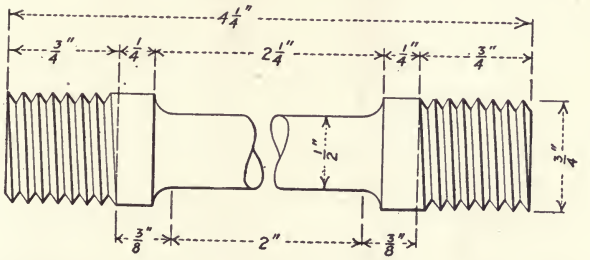

Fig. 4 Standard Form of Test Specimen Required for all Tension Tests of Steel Casting Material

welded by an approved process. The defects shall first be cleaned out to solid metal; and after welding, the castings shall be annealed, if specified by the purchaser or his representative.

c The castings offered for inspection shall not be painted or covered with any substance that will hide defects, nor rusted to such an extent as to hide defects.

V Inspection and Rejection

92 *Inspection.* The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the castings ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the castings are being furnished in accordance with

these specifications. All tests (except check analyses) and inspection shall be made at the place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

93 *Rejection.* *a* Unless otherwise specified, any rejection based on tests made in accordance with Par. 84, shall be reported within five working days from the receipt of samples.

b Castings which show injurious defects subsequent to their acceptance at the manufacturer's works will be rejected, and the manufacturer shall be notified.

94 *Rehearing.* Samples tested in accordance with Par. 84, which represent rejected castings, shall be preserved for two weeks from the date of the test report. In case of dissatisfaction with the results of the tests, the manufacturer may make claim for a rehearing within that time.

SPECIFICATIONS FOR GRAY IRON CASTINGS

THESE SPECIFICATIONS ARE IDENTICAL WITH THOSE OF THE AMERICAN SOCIETY FOR TESTING MATERIALS, SERIAL DESIGNATION A 48-05.

95 *Process of Manufacture.* Unless furnace iron is specified, all gray castings are understood to be made by the cupola process.

96 *Chemical Properties.* The sulphur contents to be as follows:

Light castings	not over 0.08 per cent
Medium castings	not over 0.10 per cent
Heavy Castings	not over 0.12 per cent

97 *Classification.* In dividing castings into light, medium and heavy classes, the following standards have been adopted:

98 Castings having any section less than ½ in. thick shall be known as *light castings.*

99 Castings in which no section is less than 2 in. thick shall be known as *heavy castings.*

100 *Medium castings* are those not included in the above classification.

Physical Properties and Tests

101 *Transverse Test.* The minimum breaking strength of the "Arbitration Bar" under transverse load shall be not under:

Light castings	2500 lbs.
Medium castings	2900 lbs.
Heavy castings	3300 lbs.

In no case shall the deflection be under 0.10 in.

102 *Tensile Test.* Where specified, this shall not run less than:

Light castings	18,000 lb. per sq. in.
Medium castings	21,000 lb. per sq. in.
Heavy castings	24,000 lb. per sq. in.

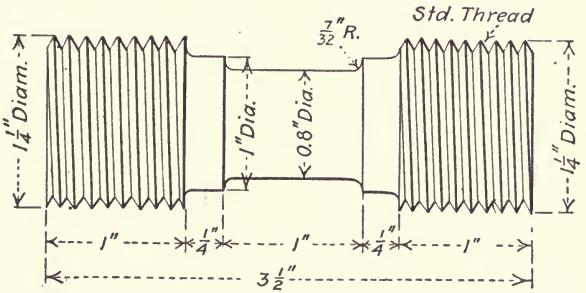

Fig. 5 Standard Form of Test Specimen Required for Tension Tests of Gray-Iron Casting Material

103 *Arbitration Bar.* The quality of the iron going into castings under specification shall be determined by means of the "Arbitration Bar." This is a bar 1¼ in. in diameter and 15 in. long. It shall be prepared as stated further on and tested transversely. The tensile test is not recommended, but in case it is called for, the bar as shown in Fig. 5, and turned up from any of the broken pieces of the transverse test shall be used. The expense of the tensile test shall fall on the purchaser.

104 *Number of Test Bars.* Two sets of two bars shall be cast from each heat, one set from the first and the other set from the last iron going into the castings. Where the heat exceeds twenty tons, an additional set of two bars shall be cast for each twenty tons or fraction thereof above this amount. In case of a change of mixture during the heat, one set of two bars shall also be cast for every mixture other

than the regular one. Each set of two bars is to go into a single mold. The bars shall not be rumbled or otherwise treated, being simply brushed off before testing.

105 *Method of Testing.* The transverse test shall be made on all the bars cast, with supports 12 in. apart, load applied at the middle,

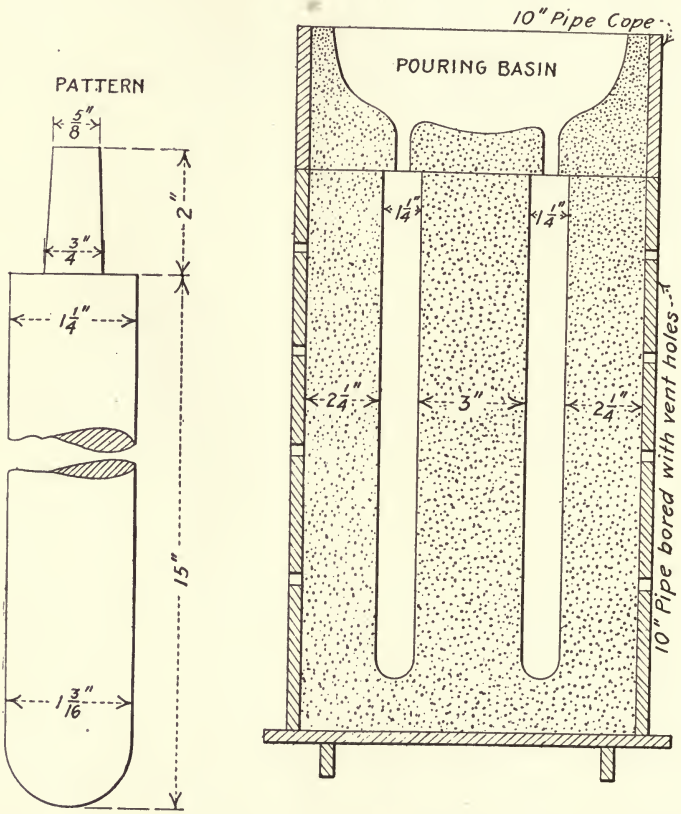

FIG. 6 DETAILS OF PATTERN AND MOLD REQUIRED FOR ARBITRATION BARS IN TESTING GRAY-IRON CASTING MATERIAL

and the deflection at rupture noted. One bar of every two of each set made must fulfill the requirements to permit acceptance of the castings represented.

106 *Mold for Test Bar.* The mold for the bars is shown in Fig. 6. The bottom of the bar is 1/16 in. smaller in diameter than the top, to allow for draft and for the strain of pouring. The pattern shall not be rapped before withdrawing. The flask is to be rammed up

with green molding sand, a little damper than usual, well mixed and put through a No. 8 sieve, with a mixture of one to twelve bituminous facing. The mold shall be rammed evenly and fairly hard, thoroughly dried and not cast until it is cold. The test bar shall not be removed from the mold until cold enough to be handled.

107 *Speed of Testing.* The rate of application of the load shall be from 20 to 40 seconds for a deflection of 0.10 in.

108 *Samples for Analysis.* Borings from the broken pieces of the "Arbitration Bar" shall be used for the sulphur determinations. One determination for each mold made shall be required. In case of dispute, the standards of the American Foundrymen's Association shall be used for comparison.

109 *Finish.* Castings shall be true to pattern, free from cracks, flaws and excessive shrinkage. In other respects they shall conform to whatever points may be specially agreed upon.

110 *Inspection.* The inspector shall have reasonable facilities afforded him by the manufacturer to satisfy him that the finished material is furnished in accordance with these specifications. All tests and inspections shall, as far as possible, be made at the place of manufacture prior to shipment.

SPECIFICATIONS FOR MALLEABLE CASTINGS

THESE SPECIFICATIONS ARE IDENTICAL WITH THOSE OF THE AMERICAN SOCIETY FOR TESTING MATERIALS, SERIAL DESIGNATION A 47-04.

111 *Process of Manufacture.* Malleable iron castings may be made by the open-hearth, air furnace, or cupola process. Cupola iron, however, is not recommended for heavy nor for important castings.

112 *Chemical Properties.* Castings for which physical requirements are specified shall not contain over 0.06 sulphur nor over 0.225 phosphorus.

Physical Properties and Tests

113 *Standard Test Bar.* This bar shall be 1 in. sq. and 14 in. long, without chills and with ends left perfectly free in the mold. Three shall be cast in one mold, heavy risers insuring sound bars. Where the full heat goes into castings which are subject to specifica-

tion, one mold shall be poured two minutes after tapping into the first ladle, and another mold from the last iron of the heat. Molds shall be suitably stamped to insure identification of the bars, the bars being annealed with the castings. Where only a partial heat is required for the work in hand, one mold should be cast from the first ladle used and another after the required iron has been tapped.

a Of the three test bars from the two molds required for each heat, one shall be tested for tensile strength and elongation, the other for transverse strength and deflection. The other remaining bar is reserved for either the transverse or tensile test, in case of the failure of the two other bars to come up to requirements. The halves of the bars broken transversely may also be used for the tensile test.

b Failure to reach the required limit for the tensile strength with elongation, as also the transverse strength with deflection, on the part of at least one test, shall reject the castings from that heat.

114 *Tensile Test.* The tensile strength of a standard test bar for castings under specification shall not be less than 40,000 lb. per sq. in. The elongation measured in 2 in. shall not be less than 2½ per cent.

115 *Transverse Test.* The transverse strength of a standard test bar, on supports 12 in. apart, pressure being applied at the center, shall not be less than 3000 lb., deflection being at least ½ in.

116 *Test Lugs.* Castings of special design or of special importance may be provided with suitable test lugs at the option of the inspector. At least one of these lugs shall be left on the casting for his inspection upon his request therefor.

117 *Annealing.* Malleable castings shall neither be "over" nor "under" annealed. They must have received their full heat in the oven at least sixty hours after reaching that temperature.

118 The "saggers" shall not be dumped until the contents shall at least be "black hot."

119 *Finish.* Castings shall be true to pattern, free from blemishes, scale or shrinkage cracks. A variation of 1/16 in. per foot shall be permissible. Founders shall not be held responsible for defects due to irregular cross sections and unevenly distributed metal.

120 *Inspection.* The inspector representing the purchaser shall have all reasonable facilities given him by the founder to satisfy him that the finished material is furnished in accordance with these specifications. All tests and inspections shall be made prior to shipment.

SPECIFICATIONS FOR BOILER RIVET IRON

THESE REQUIREMENTS ARE AN ADAPTATION, WITH SLIGHT MODIFICATIONS IN THE PHYSICAL PROPERTIES AND TESTS, OF THE SPECIFICATIONS FOR ENGINE BOLT IRON OF THE AMERICAN SOCIETY FOR TESTING MATERIALS.

A Requirements for Rolled Bars

I Manufacture

121 *Process.* The iron shall be made wholly from puddled iron or knobbled charcoal iron, and shall be free from any admixture of iron scrap or steel.

122 *Iron Scrap.* This term applies only to foreign or bought scrap and does not include local mill products free from foreign or bought scrap.

II Physical Properties and Tests

123 *Tension Tests.* *a* The iron shall conform to the following requirements as to tensile properties:

Tensile strength, lb. per sq. in.	48,000–52,000
Yield point, min., lb. per sq. in.	0.6 tens. str.
Elongation in 8 in., min., per cent.	28
Reduction of area, min., per cent.	45

b The yield point shall be determined by the drop of the beam of the testing machine. The speed of the cross-head of the machine shall not exceed 1½ in. per minute.

124 *Bend Tests.* *a* *Cold-bend Tests*—The test specimen shall bend cold through 180 deg. flat on itself without cracking on the outside of the bent portion.

b *Hot-bend Tests*—The test specimen, when heated to a bright cherry red, shall bend through 180 deg. flat on itself, without fracture on the outside of the bent portion.

c *Nick-bend Tests*—The test specimen, when nicked 25 per cent around with a tool having a 60-deg. cutting edge, to a depth of not less than 8 nor more than 16 per cent of the diameter of the specimen, and broken, shall show a wholly fibrous fracture.

d Bend tests may be made by pressure or by blows.

125 *Etch Tests.*[1] The cross-section of the test specimen shall be ground or polished, and etched for a sufficient period to develop the structure. This test shall show the material to be free from steel.

126 *Test Specimens.* All test specimens shall be of the full section of material as rolled.

127 *Number of Tests.* *a* Bars of one size shall be sorted into lots of 100 each. Two bars shall be selected at random from each lot or fraction thereof, and tested as specified in Pars. 123 and 124; but only one of these bars shall be tested as specified in Par. 125.

b If any test specimen from either of the bars originally selected to represent a lot of material, contains surface defects not visible before testing but visible after testing, or if a tension test specimen breaks outside the middle third of the gage length, one retest from a different bar will be allowed.

III Permissible Variations in Gage

128 *Permissible Variations.* The gage of each bar shall not vary more than 0.01 in. from that specified.

IV Finish

129 *Finish.* The bars shall be smoothly rolled and free from slivers, depressions, seams, crop ends and evidences of being burnt.

V Marking

130 *Marking.* The bars shall be stamped or marked as designated by the purchaser.

VI Inspection and Rejection

131 *Inspection.* *a* The inspector representing the purchaser shall have free entry at all times, while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the material ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the material is being furnished in accordance with

[1] A solution of two parts water, one part concentrated hydrochloric acid, and one part concentrated sulphuric acid is recommended for the etch test.

these specifications. Tests and inspection at the place of manufacture shall be made prior to shipment.

b The purchaser may make the tests to govern the acceptance or rejection of material in his own laboratory or elsewhere. Such tests, however, shall be made at the expense of the purchaser.

132 *Rejection.* If either of the test bars selected to represent a lot does not conform to the requirements specified in Pars. 123, 124 and 125, the lot will be rejected.

B Requirements for Rivets

I Physical Properties and Tests

133 *Number of Tests.* When specified, three rivets of each diameter shall be taken at random from each lot offered for inspection, and if they fail to stand the following tests the lot will be rejected.

134 *Bend Tests.* *a* The rivet shank shall bend cold through 180 deg. flat on itself, as shown in Fig. 2, without cracking on the outside of the bent portion.

b The heads must stand bending back, showing that they are firmly joined.

c When nicked and broken gradually the fracture must show a clean, long and fibrous iron.

II Workmanship and Finish

135 *Workmanship.* The rivets shall be true to form, concentric, and shall be made in a workmanlike manner.

136 *Finish.* The finished rivets shall be free from injurious defects.

III Inspection and Rejection

137 *Inspection.* The inspector representing the purchaser shall have free entry at all times, while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the rivets ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the rivets are being furnished in accordance with these specifications. All tests and inspection shall be made at the

place of manufacture prior to shipment, unless otherwise specified, and shall be so conducted as not to interfere unnecessarily with the operation of the works.

138 *Rejection.* Rivets which show injurious defects subsequent to their acceptance at the manufacturer's works will be rejected, and the manufacturer shall be notified.

SPECIFICATIONS FOR STAYBOLT IRON

THESE SPECIFICATIONS ARE IDENTICAL WITH THOSE OF THE AMERICAN SOCIETY FOR TESTING MATERIALS, SERIAL DESIGNATION A **39-14.**

I Manufacture

139 *Process.* The iron shall be rolled from a bloom or boxpile, made wholly from puddled iron or knobbled charcoal iron. The puddle mixture and the component parts of the bloom or boxpile shall be free from any admixture of iron scrap or steel.

140 *Definition of Terms.* *a* *Bloom*—A bloom is a solid mass of iron that has been hammered into a convenient size for rolling.

b *Boxpile*—A boxpile is a pile, the sides, top and bottom of which are formed by four flat bars and the interior of which consists of a number of small bars the full length of the pile.

c *Iron Scrap*—This term applies only to foreign or purchased scrap and does not include local mill products free from foreign or purchased scrap.

II Physcial Properties and Tests

141 *Tension Tests.* *a* The iron shall conform to the following requirements as to tensile properties:

Tensile strength, lb. per sq. in	49,000–53,000
Yield point, min., lb. per sq. in	0.6 tens. str.
Elongation in 8 in., min., per cent	30
Reduction of area, min., per cent	48

b The yield point shall be determined by the drop of the beam of the testing machine. The speed of the cross-head of the machine shall not exceed 1½ in. per minute.

142 *Bend Tests.* *a* *Cold-bend Tests*—The test specimen shall bend cold through 180 deg. flat on itself in both directions without fracture on the outside of the bent portion.

b *Quench-bend Tests*—The test specimen, when heated to a yellow heat and quenched at once in water the temperature of which is between 80 deg. and 90 deg. fahr., shall bend through 180 deg. flat on itself without fracture on the outside of the bent portion.

c *Nick-bend Tests*—The test specimen, when nicked 25 per cent around with a tool having a 60-deg. cutting edge, to a depth of not less than 8 nor more than 16 per cent of the diameter of the specimen, and broken, shall show a clean fiber entirely free from crystallization.

d Bend tests may be made by pressure or by blows.

143 *Etch Tests.*[1] The cross-section of the test specimen shall be ground or polished, and etched for a sufficient period to develop the structure. This test shall show the material to have been rolled from a bloom or a boxpile, and to be free from steel.

144 *Test Specimens.* All test specimens shall be of the full section of material as rolled.

145 *Number of Tests.* *a* Bars of one size shall be sorted into lots of 100 each. Two bars shall be selected at random from each lot or fraction thereof, and tested as specified in Pars. 141 and 142; but only one of these bars shall be tested as specified in Par. 143.

b If any test specimen from either of the bars originally selected to represent a lot of material, contains surface defects not visible before testing but visible after testing, or if a tension test specimen breaks outside the middle third of the gage length, one retest from a different bar will be allowed.

c When retests as specified in *b* are not permitted, a reduction of 2 per cent in elongation and 3 per cent in reduction of area from that specified in Par. 141, shall be allowed.

III Permissible Variations in Gage

146 *Permissible Variations.* The bars shall be truly round within 0.01 in., and shall not vary more than 0.005 in. above or more than 0.01 in. below the specified size.

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[1] A solution of two parts water, one part concentrated hydrochloric acid, and one part concentrated sulphuric acid is recommended for the etch test.

IV Finish

147 *Finish.* The bars shall be smoothly rolled and free from slivers, depressions, seams, crop ends and evidences of being burnt.

V Marking

148 *Marking.* The bars shall be stamped or marked as designated by the purchaser.

VI Inspection and Rejection

149 *Inspection. a* The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the material ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the material is being furnished in accordance with these specifications. Tests and inspection at the place of manufacture shall be made prior to shipment.

b The purchaser may make the tests to govern the acceptance or rejection of material in his own laboratory or elsewhere. Such tests, however, shall be made at the expense of the purchaser.

150 *Rejection. a* If either of the test bars selected to represent a lot does not conform to the requirements specified in Pars. 141, 142 and 143, the lot will be rejected.

b Bars which will not take a clean, sharp thread with dies in fair condition, or which develop defects in forging or machining, will be rejected, and the manufacturer shall be notified.

SPECIFICATIONS FOR REFINED WROUGHT-IRON BARS

THESE SPECIFICATIONS ARE SIMILAR TO THOSE OF THE AMERICAN SOCIETY FOR TESTING MATERIALS, SERIAL DESIGNATION A **41-13.**

I Manufacture

151 *Process.* Refined wrought-iron bars shall be made wholly from puddled iron, and may consist either of new muck-bar iron or a mixture of muck-bar iron and scrap, but shall be free from any admixture of steel.

II Physical Properties and Tests

152 *Tension Tests. a* The iron shall conform to the following minimum requirements as to tensile properties.

Tensile strength, lb. per sq. in. (See Pars. 153 and 154.)	48,000
Yield point, lb. per sq. in.	25,000
Elongation in 8 in., per cent. (See Par. 155.)	22

b The yield point shall be determined by the drop of the beam of the testing machine. The speed of the cross-head of the machine shall not exceed 1½ in. per minute.

153 *Permissible Variations.* Twenty per cent of the test specimens representing one size may show tensile strengths 1000 lb. per sq. in. under, or 5000 lb. per sq. in. over that specified in Par. 152; but no specimen shall show a tensile strength under 45,000 lb. per sq. in.

154 *Modifications in Tensile Strength.* For flat bars which have to be reduced in width, a deduction of 1000 lb. per sq. in. from the tensile strength specified in Pars. 152 and 153, shall be made.

155 *Permissible Variations in Elongation.* Twenty per cent of the test specimens representing one size may show the following percentages of elongation in 8 in.:

ROUND BARS

½ in. or over, tested as rolled	20 per cent
Under ½ in., tested as rolled	16 per cent
Reduced by machining	18 per cent

FLAT BARS

⅜ in. or over, tested as rolled	18 per cent
Under ⅜ in., tested as rolled	16 per cent
Reduced by machining	16 per cent

156 *Bend Tests.* *a* *Cold-bend Tests*—Cold bend tests will be made only on bars having a nominal area of 4 sq. in. or under, in which case the test specimen shall bend cold through 180 deg. without fracture on the outside of the bent portion, around a pin the diameter of which is equal to twice the diameter or thickness of the specimen.

b *Hot-bend Tests*—The test specimen, when heated to a temperature between 1700 deg. and 1800 deg. fahr., shall bend through 180 deg. without fracture on the outside of the bent portion, as follows: for round bars under 2 sq. in. in section, flat on itself; for round bars 2 sq. in. or over in section and for all flat bars, around a pin the diameter of which is equal to the diameter or thickness of the specimen.

c *Nick-bend Tests*—The test specimen, when nicked 25 per cent around for round bars, and along one side for flat bars, with a tool having a 60-deg. cutting edge, to a depth of not less than 8 nor more than 16 per cent of the diameter or thickness of the specimen, and broken, shall not show more than 10 per cent of the fracture surface to be crystalline.

d Bend tests may be made by pressure or by blows.

157 *Etch Tests.*[1] The cross-section of the test specimen shall be ground or polished, and etched for a sufficient period to develop the structure. This test shall show the material to be free from steel.

158 *Test Specimens.* *a* Tension and bend test specimens shall be of the full section of material as rolled, if possible; otherwise the specimens shall be machined from the material as rolled. The axis of the specimen shall be located at any point one-half the distance from the center to the surface of round bars, or from the center to the edge of flat bars, and shall be parallel to the axis of the bar.

b Etch test specimens shall be of the full section of material as rolled.

159 *Number of Tests.* *a* All bars of one size shall be piled separately. One bar from each 100 or fraction thereof will be selected at random and tested as specified.

b If any test specimen from the bar originally selected to represent a lot of material contains surface defects not visible before test-

[1] A solution of two parts water, one part concentrated hydrochloric acid, and one part concentrated sulphuric acid is recommended for the etch test.

ing but visible after testing, or if a tension test specimen breaks outside the middle third of the gage length, one retest from a different bar will be allowed.

III Permissible Variations in Gage

160 *Permissible Variations.* *a* Round bars shall conform to the standard limit gages adopted by the Master Car Builders' Association given in Table 2.

TABLE 2 PERMISSIBLE VARIATIONS IN GAGE FOR ROUND WROUGHT-IRON BARS

Nominal Diameter, Inches	Maximum Diameter, Inches	Minimum Diameter, Inches	Total Variation, Inches
¼	0.2550	0.2450	0.010
5/16	0.3180	0.3070	0.011
⅜	0.3810	0.3690	0.012
7/16	0.4440	0.4310	0.013
½	0.5070	0.4930	0.014
9/16	0.5700	0.5550	0.015
⅝	0.6330	0.6170	0.016
¾	0.7585	0.7415	0.017
⅞	0.8840	0.8660	0.018
1	1.0095	0.9905	0.019
1⅛	1.1350	1.1150	0.020
1¼	1.2605	1.2395	0.021

b The widths or thicknesses of flat bars shall not vary more than 2 per cent from that specified.

IV Finish

161 *Finish.* The bars shall be smoothly rolled and free from slivers, depressions, seams, crop ends and evidences of being burnt.

V Inspection and Rejection

162 *Inspection.* *a* The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the material ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy him that the material is being furnished in accordance with these specifications. Tests and inspection at the place of manufacture shall be made prior to shipment.

b The purchaser may make the tests to govern the acceptance or rejection of material in his own laboratory or elsewhere. Such tests, however, shall be made at the expense of the purchaser.

163 *Rejection.* All bars of one size will be rejected if the test specimens representing that size do not conform to the requirements specified.

SPECIFICATIONS FOR LAPWELDED AND SEAMLESS BOILER TUBES

Approved by the Boiler Tube Manufacturers of America September 25, 1914

I Manufacture

164 *Process.* *a* Lapwelded tubes shall be made of open-hearth steel or knobbled hammered charcoal iron.

b Seamless tubes shall be made of open-hearth steel.

II Chemical Properties and Tests

165 *Chemical Composition.* *a* The steel shall conform to the following requirements as to chemical composition:

Carbon	0.08–0.18 per cent
Manganese	0.30–0.50 per cent
Phosphorus	not over 0.04 per cent
Sulphur	not over 0.045 per cent

b Chemical analyses will not be required for charcoal iron tubes.

166 *Check Analyses.* *a* Analyses of two tubes in each lot of 250 (or on total order if less than 250) may be made by the purchaser which shall conform to the requirements specified in Par. 165. Drillings for analyses shall be taken from several points around each tube.

b If the analysis of only one tube does not conform to the requirements specified, analyses of two additional tubes from the same lot shall be made, each of which shall conform to the requirements specified.

III Physical Properties and Tests

167 *Flange Test.* *a* A test specimen not less than 4 in. in length shall have a flange turned over at right angles to the body of the tube

without showing cracks or flaws. This flange as measured from the outside of the tube shall be 3/8 in. wide.

b In making the flange test, the flaring tool and die block as shown in Fig. 7, may be used.

168 *Flattening Tests.* A test specimen 3 in. in length shall stand hammering flat until the inside walls are brought parallel and separated by a distance equal to three (3) times the wall thickness, without showing cracks or flaws. In the case lapwelded tubes, the test shall be made with the weld at the point of maximum bend.

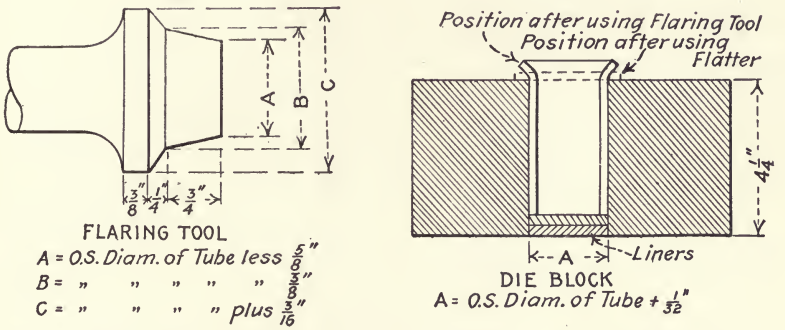

FIG. 7 DETAILS OF FLARING TOOL AND DIE BLOCK REQUIRED FOR MAKING FLANGE TESTS OF BOILER TUBES

169 *Hydrostatic Tests.* Tubes under 5 in. in diameter shall stand an internal hydrostatic pressure of 1000 lb. per sq. in. and tubes 5 in. in diameter or over, an internal hydrostatic pressure of 800 lb. per sq. in. Lapwelded tubes shall be struck near both ends, while under pressure, with a two-pound hand hammer or the equivalent.

170 *Test Specimens.* *a* All test specimens shall be taken from tubes before being cut to finished lengths and shall be smooth on the ends and free from burrs. *b* All tests shall be made cold.

171 *Number of Tests.* One flange and one flattening test shall be made from each of two tubes in each lot of 250 or less. Each tube shall be subjected to the hydrostatic test.

172 *Retests.* If the result of the physical tests of only one tube from any lot do not conform to the requirements specified in Pars. 167 and 168, retests of two additional tubes from the same lot shall be made, each of which shall conform to the requirements specified.

ETCH TESTS FOR CHARCOAL IRON

173 *Etch Tests.*[1] A cross section of tube may be turned or ground to a perfectly true surface polished free from dirt or cracks, and etched until the soft parts are sufficiently dissolved for the iron tube to show a decided ridged surface with the weld very distinct, while a steel tube would show a homogeneous surface.

IV Workmanship and Finish

174 *Workmanship.* The finished tubes shall be circular within 0.02 in. and the mean outside diameter shall not vary more than 0.015 in. from the size ordered. All tubes shall be carefully gaged with a B.W.G. gage and shall not be less than the gage specified, except the tubes on which the standard slot gage, specified, will go on tightly at the thinnest point, will be accepted. The length shall not be less, but may be 0.125 in. more than that ordered.

175 *Finish.* The finished tubes shall be free from injurious defects and shall have a workmanlike finish and shall be practically free from kinks, bends and buckles.

V Marking

176 *Marking.* The name or brand of the manufacturer, the material from which it is made, whether steel or charcoal iron, and "Tested at 1000 lb." for tubes under 5 in. in diameter, or "Tested at 800 lb." for tubes 5 in. in diameter or over, shall be legibly stenciled on each tube.

VI Inspection and Rejection

177 *Inspection.* All tests and inspection shall be made at the place of manufacture. The manufacturer of boiler tubes shall furnish the purchaser of each lot of tubes a statement of the kind of material of which the tubes are made, and that the tubes have been tested and have met all the requirements of these rules. This statement shall be furnished to the manufacturer using the tubes.

178 *Rejection.* Tubes when inserted in the boiler shall stand expanding and beading without showing cracks or flaws, or opening at the weld. Tubes which fail in this manner will be rejected and the manufacturer shall be notified.

[1] A solution of two parts of water, one part concentrated hydrochloric acid, and one part concentrated sulphuric acid is recommended for the etch test.

Construction and Maximum Allowable Working Pressures for Power Boilers

179 *Maximum Allowable Working Pressure.* The maximum allowable working pressure is that at which a boiler may be operated as determined by employing the factors of safety, stresses, and dimensions designated in these Rules.

No boiler shall be operated at a higher pressure than the maximum allowable working pressure except when the safety valve or valves are blowing, at which time the maximum allowable working pressure shall not be exceeded by more than six per cent.

Wherever the term maximum allowable working pressure is used herein, it refers to gage pressure, or the pressure above the atmosphere, in pounds per square inch.

180 The maximum allowable working pressure on the shell of a boiler or drum shall be determined by the strength of the weakest course, computed from the thickness of the plate, the tensile strength stamped thereon, as provided for in Par. 36, the efficiency of the longitudinal joint, or of the ligament between the tube holes in shell or drum, (whichever is the least), the inside diameter of the course, and the factor of the safety.

$$\frac{TS \times t \times E}{R \times FS} = \text{maximum allowable working pressure, lb. per sq. in.}$$

where

TS = ultimate tensile strength stamped on shell plates, as provided for in Par. 36, lb. per sq. in.

t = minimum thickness of shell plates in weakest course, in.

E = efficiency of longitudinal joint or of ligaments between tube holes (whichever is the least)

R = inside radius of the weakest course of the shell or drum, in.

FS = factor of safety, or the ratio of the ultimate strength of the material to the allowable stress. For new constructions covered in Part I, FS in the above formula = 5.

Boiler Joints

181 *Efficiency of a Joint.* The efficiency of a joint is the ratio which the strength of the joint bears to the strength of the solid plate. In the case of a riveted joint this is determined by calculating the breaking strength of a unit section of the joint, considering each possible mode of failure separately, and dividing the lowest result by the breaking strength of the solid plate of a length equal to that of the section considered. (See Appendix, Par. 410 to 416, for detailed methods and examples.)

182 The distance between the center lines of any two adjacent rows of rivets, or the "back pitch" measured at right angles to the direction of the joint, shall be at least twice the diameter of the rivets and shall also meet the following requirements:

a Where each rivet in the inner row comes midway between two rivets in the outer row, the sum of the two diagonal sections of the plate between the inner rivet and the two outer rivets shall be at least 20 per cent greater than the section of the plate between the two rivets in the outer row.

b Where two rivets in the inner row come between two rivets in the outer row, the sum of the two diagonal sections of the plate between the two inner rivets and the two rivets in the outer row shall be at least 20 per cent greater than the difference in the section of the plate between the two rivets in the outer row and the two rivets in the inner row.

183 On longitudinal joints, the distance from the centers of rivet holes to the edges of the plates, except rivet holes in the ends of butt straps, shall be not less than one and one-half times the diameter of the rivet holes.

184 *a Circumferential Joints.* The strength of circumferential joints of boilers, the heads of which are not stayed by tubes or through braces shall be at least 50 per cent that of the longitudinal joints of the same structure.

b When 50 per cent or more of the load which would act on an unstayed solid head of the same diameter as the shell, is relieved by the effect of tubes or through stays, in consequence of the reduction of the area acted on by the pressure and the holding power of the tubes and stays, the strength of the circumferential joints in the shell shall be at least 35 per cent that of the longitudinal joints.

185 When shell plates exceed 9/16 in. in thickness in horizontal

return tubular boilers, the portion of the plates forming the laps of the circumferential joints, where exposed to the fire or products of combustion, shall be planed or milled down as shown in Fig. 8, to ½ in. in thickness, provided the requirement in Par. 184 is complied with.

186 *Welded Joints.* The ultimate tensile strength of a longitudinal joint which has been properly welded by the forging process, shall be taken as 28,500 lb. per sq. in., with steel plates having a range in tensile strength of 47,000 to 55,000 lb. per sq. in.

187 *Longitudinal Joints.* The longitudinal joints of a shell or drum which exceeds 36 in. in diameter, shall be of butt and double-strap construction.

188 The longitudinal joints of a shell or drum which does not

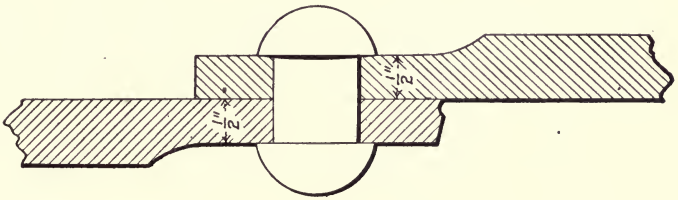

FIG. 8 CIRCUMFERENTIAL JOINT FOR THICK PLATES OF HORIZONTAL RETURN TUBULAR BOILERS

exceed 36 in. in diameter, may be of lap-riveted construction; but the maximum allowable working pressure shall not exceed 100 lb. per sq. in.

189 The longitudinal joints of horizontal return tubular boilers shall be located above the fire-line of the setting.

190 A horizontal return tubular boiler on which a longitudinal lap joint is permitted shall not have a course over 12 ft. in length. With butt and double-strap construction, longitudinal joints of any length may be used provided the plates are tested transversely to the direction of rolling, which tests shall show the standards prescribed under the Specifications of Boiler Plate Steel.

191 Butt straps and the ends of shell plates forming the longitudinal joints shall be rolled or formed by pressure, not blows, to the proper curvature.

Ligaments

192 *Efficiency of Ligament.* When a shell or drum is drilled for tubes in a line parallel to the axis of the shell or drum, the efficiency of the ligament between the tube holes shall be determined as follows:

a When the pitch of the tube holes on every row is equal (Fig. 9), the formula is:

$$\frac{p-d}{p} = \text{efficiency of ligament}$$

where

p = pitch of tube holes, in.
d = diameter of tube holes, in.

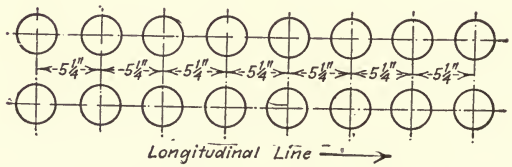

Fig. 9 Example of Tube Spacing with Pitch of Holes Equal in Every Row

Example: Pitch of tube holes in the drum as shown in Fig. 9 = 5¼ in. Diameter of tubes = 3¼ in. Diameter of tube holes = 3 9/32 in.

$$\frac{p-d}{p} = \frac{5.25-3.281}{5.25} = 0.375, \text{ efficiency of ligament}$$

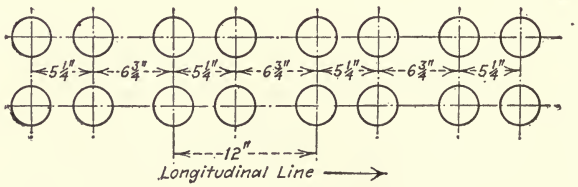

Fig. 10 Example of Tube Spacing with Pitch of Holes Unequal in Every Second Row

b When the pitch of tube holes on any one row is unequal (as in Figs. 10 or 11), the formula is:

$$\frac{p-n\,d}{p} = \text{efficiency of ligament}$$

where

p = unit length of ligament, in.
n = number of tube holes in length, p.
d = diameter of tube holes, in.

Example: Spacing shown in Fig. 10. Diameter of tube holes = 3 9/32 in.

$$\frac{p—n\,d}{p} = \frac{12—2\times 3.281}{12} = 0.453, \text{ efficiency of ligament}$$

Example: Spacing shown in Fig. 11. Diameter of tube holes = 3 9/32 in.

$$\frac{p—n\,d}{p} = \frac{29.25—5\times 3.281}{29.25} = 0.439, \text{ efficiency of ligament}$$

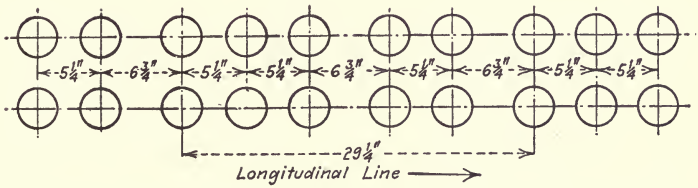

FIG. 11 EXAMPLE OF TUBE SPACING WITH PITCH OF HOLES VARYING IN EVERY SECOND AND THIRD ROW

193 When a shell or drum is drilled for tube holes in a line diagonal with the axis of the shell or drum as shown in Fig. 12, the efficiency of the ligament between the tube holes shall be determined by the following methods and the lowest value used.

a $\dfrac{0.95(p_1—d)}{p_1}$ = efficiency of ligament

b $\dfrac{p—d}{p}$ = efficiency of ligament

where

p_1 = diagonal pitch of tube holes, in.
d = diameter of tube holes, in.
p = longitudinal pitch of tube holes or distance between centers of tubes in a longitudinal row, in.

The constant 0.95 in formula *a* applies provided $\frac{p_1}{d}$ is 1.5 or over.

Example: Diagonal pitch of tube holes in drum as shown in Fig. 12 = 6.42 in.

Diameter of tube holes = 4 1/32 in.

Longitudinal pitch of tube holes = 11½ in.

$$a \qquad \frac{0.95(6.42—4.031)}{6.42} = 0.353, \text{ efficiency of ligament}$$

$$b \qquad \frac{11.5—4.031}{11.5} = 0.649, \text{ efficiency of ligament}$$

The value determined by formula *a* is the least and is the one that shall be used in this case.

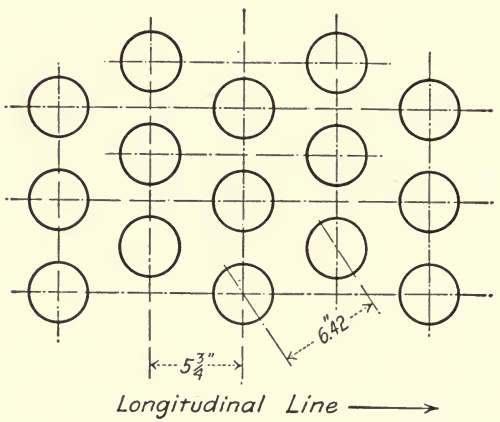

FIG. 12 EXAMPLE OF TUBE SPACING WITH TUBE HOLES ON DIAGONAL LINES

194 *Domes.* The longitudinal joint of a dome 24 in. or over in diameter shall be of butt and double-strap construction, and its flange shall be double riveted to the boiler shell when the maximum allowable working pressure exceeds 100 lb. per sq. in.

The longitudinal joint of a dome less than 24 in. in diameter may be of the lap type, and its flange may be single riveted to the boiler shell provided the maximum allowable working pressure on such a dome is computed with a factor of safety of not less than 8.

The dome may be located on the barrel or over the fire-box on traction, portable or stationary boilers of the locomotive type up to and including 48 in. barrel diameter. For larger barrel diameters, the dome shall be placed on the barrel.

Dished Heads

195 *Convex Heads.* The thickness required in an unstayed dished head with the pressure on the concave side, when it is a segment of a sphere, shall be calculated by the following formula:

$$t = \frac{5.5 \times P \times L}{2 \times TS} + \frac{1}{8}$$

where

t = thickness of plate, in.
P = maximum allowable working pressure, lb. per sq. in.
TS = tensile strength, lb. per sq. in.
L = radius to which the head is dished, in.

Where the radius is less than 80 per cent of the diameter of the shell or drum to which the head is attached the thickness shall be at least that found by the formula by making L equal to 80 per cent of the diameter of the shell or drum.

Concave Heads. Dished heads with the pressure on the convex side shall have a maximum allowable working pressure equal to 60 per cent of that for heads of the same dimensions with the pressure on the concave side.

When a dished head has a manhole opening, the thickness as found by these Rules shall be increased by not less than ⅛ in.

196 When dished heads are of a less thickness than called for by Par. 195, they shall be stayed as flat surfaces, no allowance being made in such staying for the holding power due to the spherical form.

197 The corner radius of an unstayed dished head measured on the concave side of the head shall not be less than 1½ in. or more than 4 in. and within these limits shall be not less than 3 per cent of L in Par. 195.

198 A manhole opening in a dished head shall be flanged to a depth of not less than three times the thickness of the head measured from the outside.

Braced and Stayed Surfaces

199 The maximum allowable working pressure for various thicknesses of braced and stayed flat plates and those which by these Rules require staying as flat surfaces with braces or staybolts of uni-

form diameter symmetrically spaced, shall be calculated by the formula:

$$P = C \times \frac{t^2}{p^2}$$

where

P = maximum allowable working pressure, lb. per sq. in.

t = thickness of plate in *sixteenths* of an inch

P = maximum pitch measured between straight lines passing through the centers of the staybolts in the different rows, which lines may be horizontal, vertical or inclined, in.

$C = 112$ for stays screwed through plates not over 7/16 in. thick with ends riveted over

$C = 120$ for stays screwed through plates over 7/16 in. thick with ends riveted over

$C = 135$ for stays screwed through plates and fitted with single nuts outside of plate

$C = 175$ for stays fitted with inside and outside nuts and outside washers where the diameter of washers is not less than $0.4p$ and thickness not less than t.

If flat plates not less than 3/8 in. thick are strengthened with doubling plates securely riveted thereto and having a thickness of not less than $2/3\ t$, nor more than t, then the value of t in the formula shall be 3/4 of the combined thickness of the plates and the values of C given above may also be increased 15 per cent.

200 *Staybolts.* The ends of screwed staybolts shall be riveted over or upset by equivalent process. The outside ends of such staybolts shall be drilled with a hole at least 3/16 in. diameter to a depth extending 1/2 in. beyond the inside of the plates, except on boilers having a grate area not exceeding 15 sq. ft., where the drilling of the staybolts is optional.

201 When channel irons or other members are securely riveted to the boiler heads for attaching through stays the transverse stress on such members shall not exceed 12,500 lb. per sq. in. In computing the stress, the section modulus of the member shall be used without addition for the strength of the plate. The spacing of the rivets over the supported surface shall be in conformity with that specified for staybolts.

202 The ends of stays fitted with nuts shall not be exposed to the direct radiant heat of the fire.

203 The maximum spacing between centers of rivets attaching the crowfeet of braces to the braced surface, shall be determined by the formula in Par. 199, using 135 for value of C.

The maximum spacing between the inner surface of the shell and lines parallel to the surface of the shell passing through the centers of the rivets attaching the crowfeet of braces to the head, shall be determined by the formula in Par. 199, using 160 for the value of C.

TABLE 3 MAXIMUM ALLOWABLE PITCH, IN INCHES, OF SCREWED STAYBOLTS, ENDS RIVETED OVER

Pressure, Lb. per Sq. In.	Thickness of Plate, In.						
	5/16	3/8	7/16	1/2	9/16	5/8	11/16
	Maximum Pitch of Staybolts, In.						
100	5¼	6⅜	7⅜	.....	.....	.....	.....
110	5	6	7	8⅜	.....	.....	.....
120	4¾	5¾	6¾	8	.....	.....	.....
125	4¾	5⅝	6⅝	7¾	.....	.....	.....
130	4⅝	5½	6½	7⅝	.....	.....	.....
140	4½	5⅜	6¼	7⅜	8⅜	.....	.....
150	4¼	5⅛	6	7⅛	8	.....	.....
160	4⅛	5	5⅞	6⅞	7¾	.....	.....
170	4	4⅞	5⅝	6¾	7½	8⅜	.....
180	.....	4¾	5½	6½	7⅜	8⅛	.....
190	.....	4⅝	5⅜	6⅜	7⅛	7⅞	.....
200	.....	4½	5¼	6⅛	7	7¾	8½
225	.....	4¼	4⅞	5⅞	6½	7¼	8
250	.....	4	4⅝	5½	6¼	6⅞	7⅝
300	.....	.....	4¼	5	5⅝	6¼	7

204 The formula in Par. 199 was used in computing Table 3. Where values for screwed stays with ends riveted over are required for conditions not given in Table 3, they may be computed from the formula and used, provided the pitch does not exceed 8½ in.

205 The distance from the edge of a staybolt hole to a straight line tangent to the edges of the rivet holes may be substituted for p for staybolts adjacent to the riveted edges bounding a stayed surface. When the edge of a stayed plate is flanged, p shall be measured from the inner surface of the flange, at about the line of rivets to the edge of the staybolts or to the projected edge of the staybolts.

206 The distance between the edges of the staybolt holes may be substituted for p for staybolts adjacent to a furnace door or other boiler fitting, tube hole, hand hole or other opening.

207 In water leg boilers, the staybolts may be spaced at greater distances between the rows than indicated in Table 3, provided the portions of the sheet which come between the rows of staybolts have the proper transverse strength to give a factor of safety of at least 5 at the maximum allowable working pressure.

208 The diameter of a screw stay shall be taken at the bottom of the thread, provided this is the least diameter.

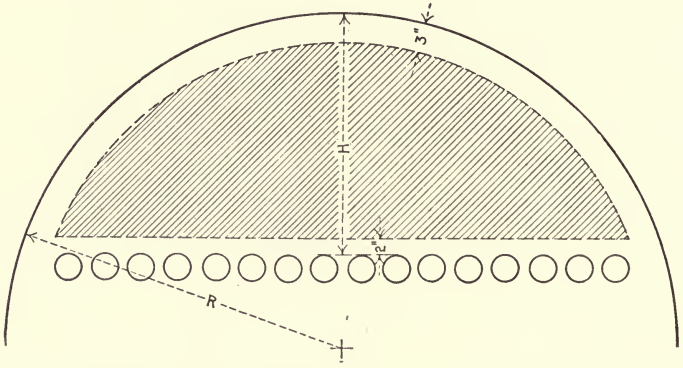

FIG. 13 METHOD OF DETERMINING NET AREA OF SEGMENT OF A HEAD

209 The least cross-sectional area of a stay shall be taken in calculating the allowable stress, except that when the stays are welded and have a larger cross-sectional area at the weld than at some other point, in which case the strength at the weld shall be computed as well as in the solid part and the lower value used.

210 Holes for screw stays shall be drilled full size or punched not to exceed 1/4 in. less than full diameter of the hole for plates over 5/16 in. in thickness, and 1/8 in. less than the full diameter of the hole for plates not exceeding 5/16 in. in thickness, and then drilled or reamed to the full diameter. The holes shall be tapped fair and true, with a full thread.

211 The ends of steel stays upset for threading, shall be thoroughly annealed.

212 An internal cylindrical furnace which requires staying shall be stayed as a flat surface as indicated in Table 3.

213 *Staying Segments of Heads.* A segment of a head shall be stayed by head to head, through, diagonal, crowfoot or gusset stays, except that a horizontal return tubular boiler may be stayed as provided in Pars. 225 to 229.

214 *Areas of Segments of Heads to be Stayed.* The area of a segment of a head to be stayed shall be the area enclosed by lines drawn 3 in. from the shell and 2 in. from the tubes, as shown in Figs. 13 and 14.

215 In water tube boilers, the tubes of which are connected to drum heads, the area to be stayed shall be taken as the total area of the head less a 5 in. annular ring, measured from the inner circumference of the drum shell.

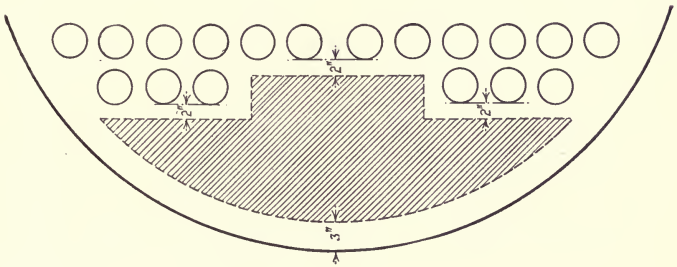

FIG. 14 METHOD OF DETERMINING NET AREA OF IRREGULAR SEGMENT OF A HEAD

When such drum heads are 30 in. or less in diameter and the tube plate is stiffened by flanged ribs or gussets, no stays need be used if a hydrostatic test to destruction of a boiler or unit section built in accordance with the construction, shows that the factor of safety is at least 5.

216 In a fire tube boiler, stays shall be used in the tube sheets if the distances between the edges of the tube holes exceed the maximum pitch of staybolts given in Table 3. That part of the tube sheet which comes between the tubes and the shell, need not be stayed when the distance from the inside of the shell to the outer surface of the tubes does not exceed that given by the formula in Par. 199, using 160 for the value of C.

217 The net area to be stayed in a segment of a head may be determined by the following formula:

$$\frac{4\ (H-5)^2}{3}\sqrt{\frac{2\ (R-3)}{(H-5)}} - 0.608 = \text{area to be stayed, sq. in.}$$

where

H = distance from tubes to shell, in.
R = radius of boiler head, in.

218 When the portion of the head below the tubes in a horizontal return tubular boiler is provided with a manhole opening, the flange of which is formed from the solid plate and turned inward to a depth of not less than three times the thickness of the head, measured from the outside, the area to be stayed as indicated in Fig. 14, may be reduced by 100 sq. in. The surface around the manhole shall be supported by through stays with nuts inside and outside at the front head.

TABLE 4 MAXIMUM ALLOWABLE STRESSES FOR STAYS AND STAYBOLTS

Description of Stays	Stresses, Lb. per Sq. In.	
	For Lengths between Supports not Exceeding 120 Diameters	For Lengths between Supports Exceeding 120 Diameters
a Unwelded stays less than twenty diameters long screwed through plates with ends riveted over	7500	....
b Unwelded stays and unwelded portions of welded stays, except as specified in line *a*	9500	8500
c Welded portions of stays	6000	6000

219 When stay rods are screwed through the sheets and riveted over, they shall be supported at intervals not exceeding 6 ft. In boilers without manholes, stay rods over 6 ft. in length may be screwed through the sheets and fitted with nuts and washers on the outside.

220 The maximum allowable stress per square inch net cross sectional area of stays and staybolts shall be as given in Table 4.

The length of the stay between supports shall be measured from the inner faces of the stayed plates. The stresses are based on tension only. For computing stresses in diagonal stays, see Pars. 221 and 222.

221 *Stresses in Diagonal and Gusset Stays.* Multiply the area of a direct stay required to support the surface by the slant or diagonal

length of the stay; divide this product by the length of a line drawn at right angles to surface supported to center of palm of diagonal stay. The quotient will be the required area of the diagonal stay.

$$A = \frac{a \times L}{l}$$

where

A = sectional area of diagonal stay, sq. in.
a = sectional area of direct stay, sq. in.
L = length of diagonal stay, as indicated in Fig. 15, in.
l = length of line drawn at right angles to boiler head or surface supported to center of palm of diagonal stay, as indicated in Fig. 15, in.

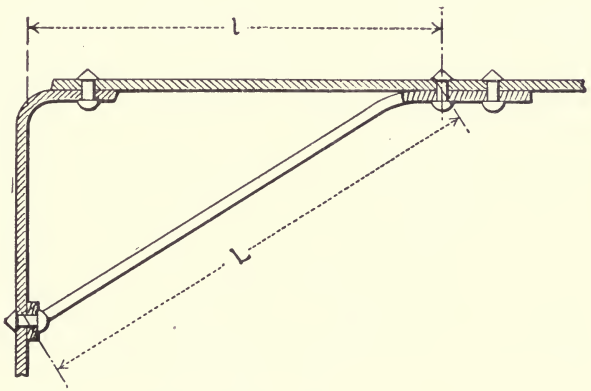

FIG. 15 MEASUREMENTS FOR DETERMINING STRESSES IN DIAGONAL STAYS

Given diameter of direct stay = 1 in., $a = 0.7854$, $L = 60$ in., $l = 48$ in.; substituting and solving:

$$A = \frac{0.7854 \times 60}{48} = 0.981 \text{ sectional area, sq. in.}$$

Diameter = 1.11 in. = 1⅛ in.

222 For staying segments of tube sheets such as in horizontal return tubular boilers, where L is not more than 1.15 times l for any brace, the stays may be calculated as direct stays, allowing 90 per cent of the stress given in Table 4.

223 *Diameter of Pins and Area of Rivets in Brace.* The sectional area of pins to resist double shear and bending when secured in crowfoot, sling, and similar stays shall be at least equal to three-

fourths of the required cross-sectional area of the brace. The combined cross section of the eye at the sides of the pin shall be at least 25 per cent greater than the required cross-sectional area of the brace.

The cross-sectional area of the rivets attaching a brace to the shell or head shall be not less than one and one quarter times the required sectional area of the brace. Each branch of a crowfoot shall be designed to carry two-thirds of the total load on the brace. The net sectional areas through the sides of the crowfeet, tee irons or similar fastenings at the rivet holes shall be at least equal to the required rivet section. All rivet holes shall be drilled and burrs removed, and the pins shall be made a neat fit.

TABLE 5 SIZES OF ANGLES REQUIRED FOR STAYING SEGMENTS OF HEADS

With the short legs of the angles attached to the head of the boiler

Height of Segment, Dimension B in Fig. 16	30″ Boiler			34″ Boiler			36″ Boiler			Dimension *A* in Fig. 16
	Angle 3″x2½″	Angle 3½″x3″	Angle 4″x3″	Angle 3½″x3″	Angle 4″x3″	Angle 5″x3″	Angle 4″x3″	Angle 5″x3″	Angle 6″x3½″	
	Thickness, inches	Thickness, inches	Thickness, inches	Thickness, inches	Thickness, inches	Thickness, inches	Thickness, inches	Thickness, inches	Thickness, inches	
10	3/8	5/16	5/16	—	—	—	—	—	—	6½
11	7/16	3/8	7/16	7/16	5/16	5/16	—	—	—	7
12	7/16	7/16	3/8	½	7/16	5/16	7/16	5/16	—	7½
13	—	7/16	7/16	11/16	½	7/16	7/16	3/8	—	8
14	—	—	½	—	5/8	3/8	5/8	7/16	3/8	8½
15	—	—	—	—	—	½	3/4	½	3/8	9
16	—	—	—	—	—	—	—	5/8	7/16	9½

224 Gusset stays when constructed of triangular right-angled web plates secured to single or double angle bars along the two sides at right angles shall have a cross-sectional area (in a plane at right angles to the longest side and passing through the intersection of the two shorter sides) not less than 10 per cent greater than would be required for a diagonal stay to support the same surface, figured by the formula in Par. 221, assuming the diagonal stay is at the same angle as the longest side of the gusset plate.

225 *Staying of Upper Segments of Tube Heads by Steel Angles.* When the shell of a boiler does not exceed 36 in. in diameter and is designed for a maximum allowable working pressure not exceeding 100 lb. per sq. in., the segment of heads above the tubes *may* be stayed by steel angles as specified in Table 5 and Fig. 16, except that angles of

equal thickness and greater depth of outstanding leg, or of greater thickness and the same depth of outstanding leg, may be substituted for those specified. The legs attached to the heads may vary in depth ½ in. above or below the dimensions specified in Table 5.

226 When this form of bracing is to be placed on a boiler, the diameter of which is intermediate to or below the diameters given in Table 5, the tabular values for the next higher diameter shall govern. Rivets of the same diameter as used in the longitudinal seams of the boiler shall be used to attach the angles to the head and to connect the outstanding legs.

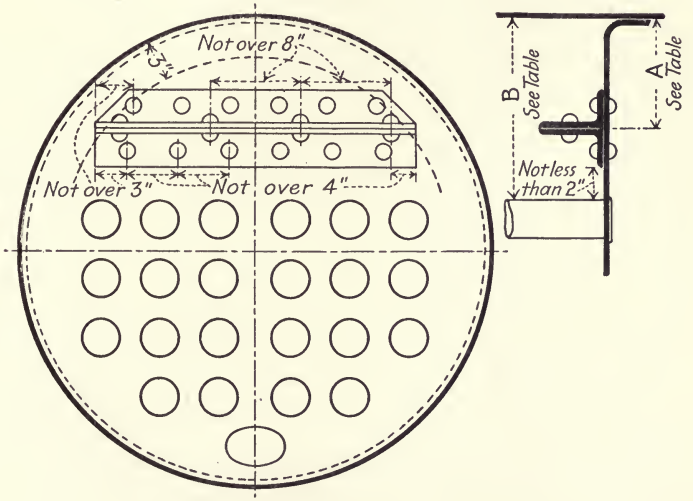

FIG. 16 STAYING OF HEAD WITH STEEL ANGLES IN TUBULAR BOILER

227 The rivets attaching angles to heads shall be spaced not over 4 in. apart. The centers of the end rivets shall be not over 3 in. from the ends of the angle. The rivets through the outstanding legs shall be spaced not over 8 in. apart; the centers of the end rivets shall be not more than 4 in. from the ends of the angles. The ends of the angles shall be considered those of the outstanding legs and the lengths shall be such that their ends overlap a circle 3 in. inside the inner surface of the shell as shown in Fig. 16.

228 The distance from the center of the angles to the shell of the boiler, marked *A* in Fig. 16, shall not exceed the values given in Table 5, but in no case shall the leg attached to the head on the lower angle come closer than 2 in. to the top of the tubes.

229 When segments are beyond the range specified in Table 5, the heads shall be braced or stayed in accordance with the requirements in these Rules.

230 *Crown Bars and Girder Stays.* Crown bars and girder stays for tops of combustion chambers and back connections, or wherever used, shall be proportioned to conform to the following formula:

$$\text{Maximum allowable working pressure} = \frac{C \times d^2 \times T}{(W - P) \times D \times W}$$

where

W = extreme distance between supports, in.
P = pitch of supporting bolts, in.
D = distance between girders from center to center, in.
d = depth of girder, in.
T = thickness of girder, in.
C = 7000 when the girder is fitted with one supporting bolt
C = 10,000 when the girder is fitted with two or three supporting bolts
C = 11,000 when the girder is fitted with four or five supporting bolts
C = 11,500 when the girder is fitted with six or seven supporting bolts
C = 12,000 when the girder is fitted with eight or more supporting bolts

Example: Given $W = 34$ in., $P = 7.5$ in., $D = 7.75$ in., $d = 7.5$ in., $T = 2$ in.; three stays per girder, $C = 10{,}000$; then substituting in formula:

Maximum allowable working pressure =

$$\frac{10{,}000 \times 7.5 \times 7.5 \times 2}{(34 - 7.5) \times 7.75 \times 34} = 161.1 \text{ lb. per sq. in.}$$

231 *Maximum Allowable Working Pressure on Truncated Cones.* Upper combustion chambers or vertical submerged tubular boilers made in the shape of a frustum of a cone when not over 38 in. diameter at the large end, may be used without stays if figured by the rule for plain cylindrical furnaces (Par. 239) making D in the formula equal to the diameter at the large end. When over 38 in. in diameter, that portion over 30 in. in diameter shall be fully supported by staybolts or gussets to conform to the provisions for the staying of flat surfaces.

232 *Stay Tubes.* When stay tubes are used in multitubular

boilers to give support to the tube plates, the sectional area of such stay tubes may be determined as follows:

$$\text{Total section of stay tubes, sq. in.} = \frac{(A-a)\ P}{T}$$

where

A = area of that portion of the tube plate containing the tubes, sq. in.

a = aggregate area of holes in the tube plate, sq. in.

P = maximum allowable working pressure, lb. per sq. in.

T = working tensile stress allowed in the tubes, not to exceed 7000 lb. per sq. in.

233 The pitch of stay tubes shall conform to the formula given in Par. 199, using the values of C as given in Table 6.

TABLE 6. VALUES OF C FOR DETERMINING PITCH OF STAY TUBES.

Pitch of Stay Tubes in the Bounding Rows	When tubes have no Nuts Outside of Plates	When tubes are Fitted with Nuts Outside of Plates
Where there are two plain tubes between each stay tube....	120	130
Where there is one plain tube between each stay tube.....	140	150
Where every tube in the bounding rows is a stay tube and each alternate tube has a nut.....	....	170

When the ends of tubes are not shielded from the action of flame or radiant heat, the values of C shall be reduced 20 per cent. The tubes shall project about ¼ in. at each end and be slightly flared. Stay tubes when threaded shall not be less than 3/16 in. thick at bottom of thread; nuts on stay tubes are not advised. For a nest of tubes C shall be taken as 140 and S as the mean pitch of stay tubes. For spaces between nests of tubes S shall be taken as the horizontal distance from center to center of the bounding rows of tubes and C as given in Table 6.

TUBE SHEETS OF COMBUSTION CHAMBERS

234 The maximum allowable working pressure on a tube sheet of a combustion chamber, where the crown sheet is not suspended from the shell of the boiler, shall be determined by the following formula:

$$P = \frac{(D-d)\ T \times 27{,}000}{W \times D}$$

where

P = maximum allowable working pressure, lb. per sq. in.
D = least horizontal distance between tube centers, in.
d = inside diameter of tubes, in.
T = thickness of tube plate, in.
W = distance from tube sheet to opposite combustion chamber sheet, in.

Example: Required the working pressure of a tube sheet supporting a crown sheet braced by crown bars. Horizontal distance between centers, 4⅛ in.; inside diameter of tubes, 2.782 in.; thickness of tube sheets, 11/16 in.; distance from tube sheet to opposite combustion chamber sheet, 34¼ in., measured from outside of tube plate to outside of back plate; material, steel. Substituting and solving:

$$P = \frac{(4.125 - 2.782) \times 0.6875 \times 27{,}000}{34.25 \times 4.125} = 176 \text{ lb. per sq. in.}$$

235 Sling stays may be used in place of girders in all cases covered in Par. 234, provided, however, that when such sling stays are

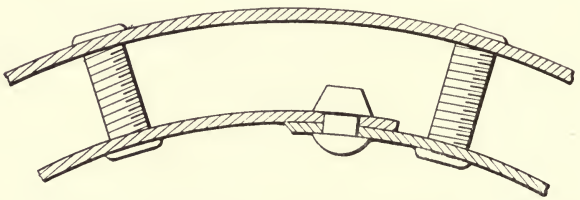

FIG. 17 PROPER LOCATION OF STAYBOLTS ADJACENT TO LONGITUDINAL JOINT IN FURNACE SHEET

used, girders or screw stays of the same sectional area shall be used for securing the bottom of the combustion chamber to the boiler shell.

236 When girders are dispensed with and the top and bottom of combustion chambers are secured by sling stays or braces, the sectional area of such stays shall conform with the requirements of rules for stays and stayed surfaces.

237 *Furnaces of Vertical Boilers.* In a vertical fire-tube boiler the furnace length, for the purpose of calculating its strength and spacing staybolts over its surface, shall be measured from the center of rivets in the bottom of the water-leg to the center of rivets in the flange of the lower tube sheet.

238 When the longitudinal joint of the furnace sheet of a vertical fire-tube boiler is of lap-riveted construction and staybolted, a staybolt in each circular row shall be located near the longitudinal joint, as shown in Fig. 17.

239 *Plain Circular Furnaces.* The maximum allowable working pressure for unstayed, riveted, seamless or lap welded furnaces, where the length does not exceed 6 times the diameter and where the thickness is at least 5/16 in. shall be determined by one or the other of the following formulae:

a Where the length does not exceed 120 times the thickness of the plate

$$P = \frac{51.5}{D}\left\{ (18.75 \times T) - (1.03 \times L) \right\}$$

b Where the length exceeds 120 times the thickness of the plate

$$P = \frac{4250 \times T^2}{L \times D}$$

where

P = maximum allowable working pressure, lb. per sq. in.
D = outside diameter of furnace, in.
L = length of furnace, in.
T = thickness of furnace walls, in sixteenths of an inch.

Where the furnaces have riveted longitudinal joints no deduction need be made for the joint provided the efficiency of the joint is greater than $P \times D$ divided by $1{,}250 \times T$.

Example. Given a furnace 26 in. in diameter, 94 in. long and ½ in. thick. The length exceeds 120 times the thickness of the plate, hence the formula (*b*) should be used. Substituting the values in this formula:

$$P = \frac{4250 \times 8 \times 8}{94 \times 26} = 111 \text{ lb. per sq. in.}$$

240 A plain cylindrical furnace exceeding 38 in. in diameter shall be stayed in accordance with the rules governing flat surfaces.

241 *Circular Flues.* The maximum allowable working pressure for seamless or welded flues more than 5 in. in diameter and up to and including 18 in. in diameter shall be determined by one or the other of the following formulae:

a Where the thickness of the wall is less than 0.023 times the diameter

$$P = \frac{10{,}000{,}000 \times T^3}{D^3}$$

b Where the thickness of the wall is greater than 0.023 times the diameter

$$P = \frac{17{,}300 \times T}{D} - 275$$

where

P = maximum allowable working pressure, lb. per sq. in.
D = outside diameter of flue, in.
T = thickness of wall of flue, in.

c The above formulae may be applied to riveted flues of the sizes specified provided the sections are not over 3 ft. in length and provided the efficiency of the joint is greater than $P \times D$ divided by $20{,}000 \times T$.

Example. Given a flue 14 in. in diameter and 5/16 in. thick. The thickness of the wall is less than 0.023 times the diameter; hence the formula (*a*) should be used. Substituting the values in this formula:

$$P = \frac{10{,}000{,}000 \times 5/16 \times 5/16 \times 5/16}{14 \times 14 \times 14} = 110 \text{ lb. per sq. in.}$$

242 *Adamson Type.* When plain horizontal flues are made in sections not less than 18 in. in length, and not less than 5/16 in. thick:

a They shall be flanged with a radius measured on the fire side, of not less than three times the thickness of the plate, and the flat portion of the flange outside of the radius shall be at least three times the diameter of the rivet holes.

b The distance from the edge of the rivet holes to the edge of the flange shall be not less than the diameter of the rivet hole, and the diameter of the rivets before driving shall be at least ¼ in. larger than the thickness of the plate.

c The depth of the Adamson ring between the flanges shall be not less than three times the diameter of the rivet holes, and the ring shall be substantially riveted to the flanges. The fire edge of the ring shall terminate at or about the point of tangency to the curve of the flange, and the thickness of the ring shall be not less than ½ in.

The maximum allowable working pressure shall be determined by the following formula:

$$P = \frac{57.6}{D} \left\{ (18.75 \times T) - (1.03 \times L) \right\}$$

where

P = maximum allowable working pressure, lb. per sq. in.
D = outside diameter of furnace, in.
L = length of furnace section, in.
T = thickness of plate, in sixteenths of an inch.

Example. Given a furnace 44 in. in diameter, 48 in. in length, and ½ in. thick. Substituting values in formula:

$$P = \frac{57.6}{44} \left\{ (18.75 \times 8) - (1.03 \times 48) \right\}$$
$$= 1.309 \ (150 - 49.44) = 131 \text{ lb. per sq. in.}$$

243 The maximum allowable working pressure on corrugated furnaces, such as the Leeds suspension bulb, Morison, Fox, Purves, or Brown, having plain portions at the ends not exceeding 9 in. in length (except flues especially provided for) when new and practically circular, shall be computed as follows:

$$P = \frac{C \times T}{D}$$

where

P = maximum allowable working pressure, lb. per sq. in.

T = thickness, in.—not less than 5/16 in. for Leeds, Morison, Fox and Brown, and not less than 7/16 in. for Purves and other furnaces corrugated by sections not over 18 in. long.

D = mean diameter, in.

C = 17,300, a constant for *Leeds furnaces,* when corrugations are not more than 8 in. from center to center and not less than 2¼ in. deep.

C = 15,600, a constant for *Morison furnaces,* when corrugations are not less than 8 in. from center to center and the radius of the outer corrugations is not more than one half that of the suspension curve.

C = 14,000, a constant for *Fox Furnaces,* when corrugations are not more than 8 in. from center to center and not less than 1½ in. deep.

C = 14,000, a constant for *Purves furnaces* when rib projections are not more than 9 in. from center to center and not less than 1⅜ in. deep.

C = 14,000, a constant for *Brown Furnaces,* when corrugations are not more than 9 in. from center to center and not less than 1⅝ in. deep.

C = 10,000, a constant for furnaces corrugated by sections not more than 18 in. from center to center and not less than 2½ in. deep, measured from the least inside to the greatest outside diameter of the corrugations, and having the ends fitted one into the other and substantially riveted together, provided that the plain parts at the ends do not exceed 12 in. in length.

In calculating the mean diameter of the Morison furnace, the least inside diameter plus 2 in., may be taken as the mean diameter.

244 The thickness of a corrugated or ribbed furnace shall be ascertained by actual measurement. The furnace shall be drilled for a ¼-in. pipe tap and fitted with a screw-plug that can be removed for the purpose of measurement. For the Brown and Purves furnaces, the holes shall be in the center of the second flat; for the Morison, Fox and other similar types, in the center of the top corrugation, at least as far in as the fourth corrugation from the end of the furnace.

245 *Cast Iron Headers.* The pressure allowed on a water-tube boiler, the tubes of which are secured to cast-iron or malleable-iron headers, shall not exceed 160 lb. per sq. in. The form and size of the internal cross section of a cast-iron or malleable-iron header at any point shall be such that it will fall within a 6 in. by 7 in. rectangle.

246 The cast-iron used for the headers of water-tube boilers shall conform with the Specifications for Gray-iron Castings given in Pars. 95 to 110, the header to be arbitrarily classified as a "medium casting" as to physical properties and tests, and as a "light casting" as to chemical properties.

247 A cast-iron header when tested to destruction, shall withstand a hydrostatic pressure of at least 1200 lb. per sq. in. A hydrostatic test at 400 lb. per sq. in. gage pressure shall be made on all new headers with tubes attached.

Tubes

248 *Tube Holes and Ends.* Tube holes shall be drilled full size from the solid plate, or they may be punched at least ½ in. smaller in diameter than full size, and then drilled, reamed or finished full size with a rotating cutter.

249 The sharp edges of tube holes shall be taken off on both sides of the plate with a file or other tool.

250 A fire-tube boiler shall have the ends of the tubes substantially rolled and beaded, or welded at the firebox or combustion chamber end.

251 The ends of all tubes, suspension tubes and nipples shall be flared not less than ⅛ in. over the diameter of the tube hole on all water-tube boilers and superheaters, or they may be beaded.

252 The ends of all tubes, suspension tubes and nipples of water-tube boilers and superheaters shall project through the tube sheets or headers not less than ¼ in. nor more than ½ in. before flaring.

Riveting

253 *Riveting.* Rivet holes, except for attaching stays or angle bars to heads, shall be drilled full size with plates, butt straps and heads bolted in position; or they may be punched not to exceed ¼ in. less than full diameter for plates over 5/16 in. in thickness, and ⅛ in. less than full diameter for plates not exceeding 5/16 in. in thickness, and then drilled or reamed to full diameter with plates, butt straps and heads bolted in position.

254 After drilling rivet holes, the plates and butt straps shall be separated and the burrs removed.

255 *Rivets.* Rivets shall be of sufficient length to completely fill the rivet holes and form heads at least equal in strength to the bodies of the rivets.

256 Rivets shall be machine driven wherever possible, with sufficient pressure to fill the rivet holes, and shall be allowed to cool and shrink under pressure.

Calking

257 *Calking.* The calking edges of plates, butt straps and heads shall be beveled. Every portion of the calking edges of plates, butt straps and heads shall be planed, milled or chipped to a depth of not less than ⅛ in. Calking shall be done with a round-nosed tool.

Manholes

258 *Manholes.* An elliptical manhole opening shall be not less than 11 × 15 in. or 10 × 16 in. in size. A circular manhole opening shall be not less than 15 in. in diameter.

259 A manhole reinforcing ring when used, shall be of steel or wrought-iron, and shall be at least as thick as the shell plate.

260 Manhole frames on shells or drums when used, shall have the proper curvature, and on boilers over 48 in. in diameter shall be riveted to the shell or drum with two rows of rivets, which may be pitched as shown in Fig. 18. The strength of the rivets in shear on manhole frames and reinforcing rings shall be at least equal to the tensile strength of that part of the shell plate removed, on a line parallel to the axis of the shell, through the center of the manhole, or other opening.

261 The proportions of manhole frames and other reinforcing rings to conform to the above specifications may be determined by the use of the following formulae, which are based on the assumption that the rings shall have the same tensile strength per square inch of section as, and be of not less thickness than, the shell plate removed.

For a single-riveted ring: $W = \frac{l \times t_1}{2 \times t} + d$

For a double-riveted ring: $W = \frac{l \times t_1}{2 \times t} + 2d$

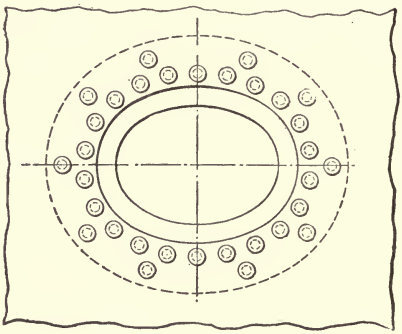

FIG. 18 METHOD OF RIVETING MANHOLE FRAMES TO SHELLS OR DRUMS WITH TWO ROWS OF RIVÉTS

For two single-riveted rings: $W = \frac{l \times t_1}{4 \times t} + d$

For two double-riveted rings: $W = \frac{l \times t_1}{4 \times t} + 2d$

Where

W = least width of reinforcing ring, in.

t_1 = thickness of shell plate, in.

d = diameter of rivet when driven, in.

t = thickness of reinforcing ring—not less than thickness of the shell plate, in.

T = tensile strength of the ring, lb. per sq. in. of section

a = net section of one side of the ring or rings, sq. in.

S = shearing strength of rivet, lb. per sq. in. of section (see Par. 16)

l = length of opening in shell in direction parallel to axis of shell, in.

N = number of rivets

To find the number of rivets for a single or double reinforcing ring:

$$N = \frac{5.1 \times T \times a}{S \times d^2}$$

262 Manhole plates shall be of wrought steel or shall be steel castings.

263 The minimum width of bearing surface, for a gasket on a manhole opening shall be ½ in. No gasket for use on a manhole or handhole of any boiler shall have a thickness greater than ¼ in.

264 A manhole shall be located in the front head, below the tubes, of a horizontal return tubular boiler 48 in. or over in diameter. Smaller boilers shall have either a manhole or a handhole below the tubes. There shall be a manhole in the upper part of the shell or head of a fire-tube boiler over 40 in. in diameter, except a vertical fire-tube boiler, or except on internally fired boilers not over 48 in. in diameter. The manhole may be placed in the head of the dome. Smaller boilers shall have either a manhole or a handhole above the tubes.

Washout Holes

265 A traction, portable or stationary boiler of the locomotive type shall have not less than six handholes, or washout plugs, located as follows: one in the rear head below the tubes; one in the front head at or about the line of the crown sheet; four in the lower part of the waterleg; also, where possible, one near the throat sheet.

266 A vertical fire-tube boiler, except the boiler of a steam fire-engine, shall have not less than seven handholes, located as follows: three in the shell at or about the line of the crown sheet; one in the shell at or about the line of the fusible plug when used; three in the shell at the lower part of the waterleg. A vertical fire-tube boiler, submerged tube type, shall have two or more handholes in the shell, in line with the upper tube sheet.

267 A vertical fire-tube boiler of a steam fire-engine shall have at least three brass washout plugs of not less than 1-in. iron pipe size, screwed into the shell and located as follows: one at or about the line of the crown sheet; two at the lower part of the waterleg.

Threaded Openings

268 *Threaded Openings.* An opening in a boiler for a threaded pipe connection 1 in. in diameter or over shall have not less than the number of threads given in Table 7.

TABLE 7 MINIMUM NUMBER OF PIPE THREADS FOR CONNECTIONS TO BOILERS

Size of pipe connection, in.	1 and 1¼	1½ and 2	2½ to 4 inclusive	4½ to 6 inclusive	7 and 8	9 and 10	12
Number of threads per in.	11½	11½	8	8	8	8	8
Minimum number of threads required in opening	4	5	7	8	10	12	13
Minimum thickness of material required to give above number of threads, in.	0.348	0.435	0.875	1	1.25	1.5	1.625

If the thickness of the material in the boiler is not sufficient to give such number of threads, there shall be a pressed steel flange, bronze composition flange, steel-cast flange or steel plate, so as to give the required number of threads, constructed and riveted to the boiler in accordance with methods given in Par. 261. A steam main or safety valve opening may be fitted with either a steel cast, wrought-steel or bronze composition nozzle. A feed-pipe connection may be fitted with a brass or steel boiler bushing.

Safety Valves

269 *Safety Valve Requirements.* Each boiler shall have two or more safety valves, except a boiler for which one safety valve 3-in. size or smaller is required by these Rules.

270 The safety valve capacity for each boiler shall be such that the safety valve or valves will discharge all the steam that can be generated by the boiler without allowing the pressure to rise more than six per cent above the maximum allowable working pressure, or more than six per cent above the highest pressure to which any valve is set.

271 One or more safety valves on every boiler shall be set at or below the maximum allowable working pressure. The remaining

valves may be set within a range of three per cent above the maximum allowable working pressure, but the range of setting of all of the valves on a boiler shall not exceed ten per cent of the highest pressure to which any valve is set.

272 Safety valves shall be of the direct spring loaded pop type with seat and bearing surface of the disc either inclined at an angle of about 45 deg. or flat at an angle of about 90 deg. to the center line of the spindle. The vertical lift of the valve disc measured immediately after the sudden lift due to the pop may be made any amount desired up to a maximum of 0.15 in. irrespective of the size of the valve. The nominal diameter measured at the inner edge of the valve seat shall be not less than 1 in. or more than 4½ in.

273 Each safety valve shall have plainly stamped or cast on the body:

a The name or identifying trade-mark of the manufacturer

b The nominal diameter with the words "Bevel Seat" or "Flat Seat"

c The steam pressure at which it is set to blow

d The lift of the valve disc from its seat, measured immediately after the sudden lift due to the pop

e The weight of steam discharged in pounds per hour at the pressure for which it is set to blow.

274 The minimum capacity of a safety valve or valves to be placed on a boiler shall be determined on the basis of 6 lb. of steam per hour per sq. ft. of boiler heating surface for water tube boilers, and 5 lb. for all other types of power boilers, and upon the relieving capacity marked on the valves by the manufacturer, provided such marked relieving capacity does not exceed that given in Table 8. In case the relieving capacity marked on the valve or valves exceeds the maximum given in Table 8, the minimum safety valve capacity shall be determined on the basis of the maximum relieving capacity given in Table 8 for the particular size of valve and working pressure for which it was constructed. The heating surface shall be computed for that side of the boiler surface exposed to the products of combustion, exclusive of the superheating surface. In computing the heating surface for this purpose only the tubes, shells, tube sheets and the projected area of headers need be considered.

TABLE 8 DISCHARGE CAPACITIES FOR DIRECT SPRING-LOADED POP SAFETY VALVES, WITH 45 DEG. BEVEL SEATS

Gage Pres., Lb. per Sq. In.		Diameter, 1 In.			Diameter, 1¼ In.			Diameter, 1½ In.		
		Min.	Int.	Max.	Min.	Int.	Max.	Min.	Int.	Max.
15	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	95,500	191,000	238,900	179,200	238,800	293,500	214,900	358,300	429,900
	Lb. hr....	65	131	163	122	163	203	146	245	293
25	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	127,700	255,400	319,300	239,500	319,300	399,100	287,400	478,900	574,700
	Lb. hr....	87	174	218	164	218	272	196	326	392
50	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	208,200	416,400	520,400	390,300	520,400	650,500	468,300	780,600	936,600
	Lb. hr....	142	284	354	266	354	444	320	532	639
75	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	288,600	577,200	721,400	541,100	721,400	901,800	649,300	1,082,000	1,299,000
	Lb. hr....	197	393	492	369	492	615	443	738	886
100	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	369,000	738,000	922,500	691,900	922,500	1,153,000	830,300	1,384,000	1,661,000
	Lb. hr....	252	503	629	472	629	786	566	944	1133
125	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	449,400	898,900	1,124,000	842,700	1,124,000	1,404,000	1,011,000	1,685,000	2,022,000
	Lb. hr....	307	613	767	575	767	957	689	1149	1379
150	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	529,900	1,060,000	1,325,000	993,500	1,325,000	1,656,000	1,192,000	1,987,000	2,384,000
	Lb. hr....	362	723	904	677	904	1129	813	1355	1625
175	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	610,300	1,221,000	1,526,000	1,144,000	1,526,000	1,907,000	1,373,000	2,289,000	2,746,000
	Lb. hr....	416	833	1040	780	1040	1301	936	1561	1873
200	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	690,700	1,381,000	1,727,000	1,295,000	1,727,000	2,158,000	1,554,000	2,590,000	3,108,000
	Lb. hr....	471	941	1178	883	1178	1472	1060	1766	2119
225	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	771,100	1,542,000	1,928,000	1,446,000	1,928,000	2,410,000	1,735,000	2,892,000	3,470,000
	Lb. hr....	526	1052	1315	986	1315	1643	1183	1972	2366
250	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	851,600	1,703,000	2,129,000	1,597,000	2,129,000	2,661,000	1,916,000	3,193,000	3,832,000
	Lb. hr....	581	1161	1451	1089	1451	1814	1307	2177	2613
275	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	932,000	1,864,000	2,330,000	1,748,000	2,330,000	2,913,000	2,097,000	3,495,000	4,194,000
	Lb. hr....	635	1271	1589	1192	1589	1986	1430	2383	2860
300	Lift, in....	0.02	0.04	0.05	0.03	0.04	0.05	0.03	0.05	0.06
	CH.....	1,024,000	2,048,000	2,531,000	1,898,000	2,531,000	3,164,000	2,278,000	3,797,000	4,556,000
	Lb. hr....	698	1397	1746	1294	1726	2157	1553	2589	3107

The Discharge capacity of a Flat Seat Valve of a given diameter with a given lift may be obtained by multiplying the discharge capacity given in the Table for a 45 deg. bevel seat valve of same diameter and same lift, by 1.4.

TABLE 8 (CONTINUED) DISCHARGE CAPACITIES FOR DIRECT SPRING-LOADED POP SAFETY VALVES, WITH 45 DEG. BEVEL SEATS

Gage Pres., Lb. per Sq. In.		Diameter, 2 In.			Diameter, 2½ In.			Diameter, 3 In.		
		Min.	Int.	Max.	Min.	Int.	Max.	Min.	Int.	Max.
15	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	382,200	573,300	668,900	477,700	716,600	955,500	716,600	1,147,000	1,433,000
	Lb. hr....	261	391	456	326	488	651	489	782	977
25	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	510,900	766,300	894,000	638,500	957,900	1,277,000	957,900	1,533,000	1,916,000
	Lb. hr....	349	523	610	435	653	871	653	1046	1307
50	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	832,600	1,249,000	1,457,000	1,041,000	1,561,000	2,081,000	1,561,000	2,498,000	3,122,000
	Lb. hr....	568	851	994	710	1064	1419	1064	1703	2129
75	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	1,154,000	1,731,000	2,020,000	1,443,000	2,164,000	2,886,000	2,164,000	3,463,000	4,329,000
	Lb. hr....	787	1181	1377	984	1475	1968	1475	2361	2951
100	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	1,476,000	2,214,000	2,583,000	1,845,000	2,768,000	3,690,000	2,768,000	4,428,000	5,535,000
	Lb. hr....	1007	1510	1761	1258	1887	2516	1887	3019	3774
125	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	1,795,000	2,693,000	3,146,000	2,247,000	3,371,000	4,494,000	3,371,000	5,393,000	6,741,000
	Lb. hr....	1224	1836	2145	1532	2299	3064	2299	3677	4596
150	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	2,109,000	3,179,000	3,709,000	2,649,000	3,974,000	5,299,000	3,974,000	6,358,000	7,948,000
	Lb. hr....	1438	2158	2529	1806	2710	3613	2710	4335	5419
175	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	2,441,000	3,662,000	4,272,000	3,051,000	4,577,000	6,103,000	4,577,000	7,323,000	9,154,000
	Lb. hr....	1664	2497	2913	2081	3121	4161	3121	4993	6242
200	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	2,763,000	4,144,000	4,835,000	3,454,000	5,180,000	6,907,000	5,180,000	8,289,000	10,361,000
	Lb. hr....	1884	2826	3296	2354	3532	4709	3532	5651	7064
225	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	3,085,000	4,626,000	5,398,000	3,856,000	5,784,000	7,711,000	5,784,000	9,254,000	11,567,000
	Lb. hr....	2104	3154	3680	2629	3944	5258	3944	6310	7890
250	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	3,406,000	5,109,000	5,961,000	4,258,000	6,387,000	8,516,000	6,387,000	10,219,000	12,774,000
	Lb. hr....	2322	3484	4064	2903	4355	5807	4355	6968	8708
275	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	3,728,000	5,592,000	6,524,000	4,660,000	6,990,000	9,320,000	6,990,000	11,180,000	13,980,000
	Lb. hr....	2542	3813	4448	3177	4766	6355	4766	7620	9533
300	Lift, in....	0.04	0.06	0.07	0.04	0.06	0.08	0.05	0.08	0.10
	CH.....	4,050,000	6,075,000	7,087,000	5,062,000	7,593,000	10,124,000	7,593,000	12,149,000	15,186,000
	Lb. hr....	2762	4143	4832	3452	5177	6903	5177	8280	10,358

The Discharge capacity of a Flat Seat Valve of a given diameter with a given lift may be obtained by multiplying the discharge capacity given in the Table for a 45 deg. bevel seat valve of same diameter and same lift, by 1.4.

This table is concluded on the following page.

TABLE 8 (Concluded) DISCHARGE CAPACITIES FOR DIRECT SPRING-LOADED POP SAFETY VALVES, WITH 45 DEG. BEVEL SEATS

Gage Pres., Lb. per Sq. In.		Diameter, 3½ In.			Diameter, 4 In.			Diameter, 4½ In.		
		Min.	Int.	Max.	Min.	Int.	Max.	Min.	Int.	Max.
15	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	1,003,000	1,505,000	1,839,000	1,338,000	1,911,000	2,293,000	1,720,000	2,365,000	2,795,000
	Lb. hr....	684	1026	1254	912	1303	1564	1173	1613	1906
25	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	1,341,000	2,012,000	2,459,000	1,788,000	2,554,000	3,065,000	2,299,000	3,161,000	3,736,000
	Lb. hr....	914	1372	1676	1219	1742	2090	1568	2156	2547
50	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	2,186,000	3,278,000	4,007,000	2,914,000	4,163,000	4,996,000	3,747,000	5,152,000	6,088,000
	Lb. hr....	1490	2235	2732	1987	2839	3406	2555	3513	4151
75	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	3,030,000	4,545,000	5,555,000	4,040,000	5,772,000	6,926,000	5,194,000	7,142,000	8,441,000
	Lb. hr....	2066	3099	3788	2754	3935	4722	3542	4870	5756
100	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	3,875,000	5,812,000	7,103,000	5,166,000	7,380,000	8,856,000	6,642,000	9,133,000	10,793,000
	Lb. hr....	2642	3963	4843	3522	5032	6038	4529	6227	7358
125	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	4,719,000	7,079,000	8,652,000	6,292,000	8,988,000	10,786,000	8,089,000	11,123,000	13,146,000
	Lb. hr....	3218	4826	5899	4290	6128	7354	5516	7583	8963
150	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	5,564,000	8,345,000	10,199,000	7,418,000	10,597,000	12,717,000	9,537,000	13,114,000	15,498,000
	Lb. hr....	3794	5690	6954	5058	7226	8670	6503	8940	10566
175	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	6,408,000	9,612,000	11,748,000	8,544,000	12,206,000	14,647,000	10,985,000	15,105,000	17,851,000
	Lb. hr....	4369	6553	8010	5824	8320	9984	7490	10298	12173
200	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	7,253,000	10,879,000	13,296,000	9,670,000	13,814,000	16,580,000	12,433,000	17,095,000	20,204,000
	Lb. hr....	4946	7418	9068	6593	9420	11305	8475	11655	13773
225	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	8,097,000	12,146,000	14,845,000	10,796,000	15,423,000	18,507,000	13,881,000	19,086,000	22,556,000
	Lb. hr....	5521	8280	10120	7361	10514	12616	9465	13013	15383
250	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	8,942,000	13,412,000	16,393,000	11,922,000	17,031,000	20,438,000	15,328,000	21,076,000	24,908,000
	Lb. hr....	6097	9143	11175	8130	11614	13938	10448	14366	16980
275	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	9,786,000	14,679,000	17,941,000	13,048,000	18,640,000	22,368,000	16,776,000	23,067,000	27,261,000
	Lb. hr....	6672	10005	12233	8895	12707	15248	11438	15728	18585
300	Lift, in....	0.06	0.09	0.11	0.07	0.10	0.12	0.08	0.11	0.13
	CH.....	10,630,000	15,946,000	19,489,000	14,174,000	20,249,000	24,298,000	18,224,000	25,058,000	29,614,000
	Lb. hr....	7248	10875	13290	9668	13807	16568	12428	17088	20195

The Discharge capacity of a Flat Seat Valve of a given diameter with a given lift may be obtained by multiplying the discharge capacity given in the Table for a 45 deg. bevel seat valve of same diameter and same lift, by 1.4.

275 Safety valve capacity may be checked in any one of three different ways, and if found sufficient, additional capacity need not be provided:

a By making an accumulation test, by shutting off all other steam discharge outlets from the boiler and forcing the fires to the maximum. The safety valve equipment shall be sufficient to prevent an excess pressure beyond six per cent as specified in Par. 270.

b By measuring the maximum amount of fuel that can be burned and computing the corresponding evaporative capacity upon the basis of the heating value of the fuel. See Appendix, Pars. 421 to 427.

c By determining the maximum evaporative capacity by measuring the feed water. The sum of the safety valve capacities marked on the valves, shall be equal to or greater than the maximum evaporative capacity of the boiler.

276 When two or more safety valves are used on a boiler, they may be either separate or twin valves made by mounting individual valves on Y-bases, or duplex, triplex or multiplex valves having two or more valves in the same body casing.

277 The safety valve or valves shall be connected to the boiler independent of any other steam connection, and attached as close as possible to the boiler, without any unnecessary intervening pipe or fitting. Every safety valve shall be connected so as to stand in an upright position, with spindle vertical, when possible.

278 Each safety valve shall have full sized direct connection to the boiler. No valve of any description shall be placed between the safety valve and the boiler, nor on the discharge pipe between the safety valve and the atmosphere. When a discharge pipe is used, it shall be not less than the full size of the valve, and shall be fitted with an open drain to prevent water from lodging in the upper part of the safety valve or in the pipe.

279 If a muffler is used on a safety valve it shall have sufficient outlet area to prevent back pressure from interfering with the proper operation and discharge capacity of the valve. The muffler plates or other devices shall be so constructed as to avoid any possibility of restriction of the steam passages due to deposit. When an elbow is placed on a safety valve discharge pipe, it shall be located close to the safety valve outlet or the pipe shall be securely anchored and supported. All safety valve discharges shall be so located or piped as to be carried

clear from running boards or working platforms used in controlling the main stop valves of boilers or steam headers.

280 When a boiler is fitted with two or more safety valves on one connection, this connection to the boiler shall have a cross-sectional area not less than the combined area of all of the safety valves with which it connects.

281 Safety valves shall operate without chattering and shall be set and adjusted as follows: To close after blowing down not more than 4 lb. on boilers carrying an allowed pressure less than 100 lb. per sq. in. gage. To close after blowing down not more than 6 lb. on boilers carrying pressures between 100 and 200 lb. per sq. in. gage inclusive. To close after blowing down not more than 8 lb. on boilers carrying over 200 lb. per sq. in. gage.

282 Each safety valve used on a boiler shall have a substantial lifting device, and shall have the spindle so attached that the valve disc can be lifted from its seat a distance not less than one-tenth of the nominal diameter of the valve, when there is no pressure on the boiler.

283 The seats and discs of safety valves shall be of non-ferrous material.

284 Springs used in safety valves shall not show a permanent set exceeding 1/32 in. ten minutes after being released from a cold compression test closing the spring solid.

285 The spring in a safety valve shall not be used for any pressure more than 10 per cent above or below that for which it was designed.

286 A safety valve over 3-in. size, used for pressures greater than 15 lb. per sq. in. gage, shall have a flanged inlet connection. The dimensions of the flanges shall conform to the American standard given in Tables 15 and 16 of the Appendix.

287 When the letters *A S M E Std* are plainly stamped or cast on the valve body this shall be a guarantee by the manufacturer that the valve conforms with the details of construction herein specified.

288 Every superheater shall have one or more safety valves near the outlet. The discharge capacity of the safety valve or valves on an attached superheater may be included in determining the number and sizes of the safety valves for the boiler, provided there are no intervening valves between the superheater safety valve and the boiler.

289 Every safety valve used on a superheater, discharging superheated steam, shall have a steel body with a flanged inlet connection,

and shall have the seat and disc of nickel composition or equivalent material, and the spring fully exposed outside of the valve casing so that it shall be protected from contact with the escaping steam.

290 Every boiler shall have proper outlet connections for the required safety valve or valves, independent of any other steam outlet connection or of any internal pipe in the steam space of the boiler, the area of opening to be at least equal to the aggregate area of all of the safety valves to be attached thereto.

Water and Steam Gages

291 *Water Glasses and Gage Cocks.* Each boiler shall have at least one water glass, the lowest visible part of which shall be not less than 2 in. above the lowest permissible water level.

292 No water glass connection shall be fitted with an automatic shut-off valve.

293 When shut-offs are used on the connections to a water column, they shall be either outside screw and yoke type gate valves or stop cocks with levers permanently fastened thereto, and such valves or cocks shall be locked or sealed *open.*

294 Each boiler shall have three or more gage cocks, located within the range of the visible length of the water glass, except when such boiler has two water glasses with independent connections to the boiler and located on the same horizontal line and not less than 2 ft. apart.

295 No outlet connections, except for damper regulator, feedwater regulator, drains or steam gages, shall be placed on the pipes connecting a water column to a boiler.

296 *Steam Gages.* Each boiler shall have a steam gage connected to the steam space or to the water column or its steam connection. The steam gage shall be connected to a syphon or equivalent device of sufficient capacity to keep the gage tube filled with water and so arranged that the gage cannot be shut off from the boiler except by a cock placed near the gage and provided with a tee or lever handle arranged to be parallel to the pipe in which it is located when the cock is open. Connections to gages shall be of brass, copper or bronze composition.

297 The dial of the steam gage shall be graduated to not less than 1½ times the maximum allowable working pressure on the boiler.

298 Each boiler shall be provided with a ¼-in. pipe size valved

connection for attaching a test gage when the boiler is in service, so that the accuracy of the boiler steam gage can be ascertained.

Fittings and Appliances

299 *Nozzles and Fittings.* All fittings shall conform to the American Standards given in Tables 15 or 16 of the Appendix. Where the maximum allowable working pressure is less than 125 lb. per sq. in., Table 15 shall be used and where higher, Table 16.

300 The minimum number of threads that a pipe or fitting shall screw into a tapped hole shall correspond to the numerical values given for number of threads in Table 7.

301 *Stop Valves.* Each steam discharge outlet over 2 in. in diameter, except safety valve and superheater connections, shall be fitted with a stop valve or valves of the outside screw and yoke type, located as near the boiler as practicable.

302 The main stop valves of boilers shall be extra heavy when the maximum allowable working pressure exceeds 125 lb. per sq. in. The fittings between the boiler and such valve or valves shall be extra heavy, as specified in Table 16 of the Appendix.

303 When two or more boilers are connected to a common steam main, two stop valves, with an ample free blow drain between them, shall be placed in the steam connection between each boiler and the steam main. The discharge of this drain valve must be visible to the operator while manipulating the valve. The stop valves shall consist preferably of one automatic non-return valve (set next the boiler) and a second valve of the outside screw and yoke type; or, two valves of the outside screw and yoke type may be used.

304 When a stop valve is so located that water can accumulate, ample drains shall be provided.

305 *Steam Mains.* Provisions shall be made for the expansion and contraction of steam mains connected to boilers, by providing substantial anchorage at suitable points, so that there shall be no undue strain transmitted to the boiler. Steam reservoirs shall be used on steam mains when heavy pulsations of the steam currents cause vibration of the boiler shell plates.

306 Each superheater shall be fitted with a drain.

307 *Blow-off Piping.* The size of a surface blow-off pipe shall not exceed 1½ in., and it shall be carried through the shell or head with a brass or steel boiler bushing.

308 Each boiler shall have a bottom blow-off pipe, fitted with a valve or cock, in direct connection with the lowest water space practicable; the minimum size of pipe and fittings shall be 1 in. and the maximum size shall be 2½ in. Globe valves shall not be used on such connections.

309 A bottom blow-off cock shall have the plug held in place by a guard or gland. The end of the plug shall be distinctly marked in line with the passage.

310 The blow-off pipe or pipes shall be extra heavy from boiler to valve or valves, and shall run full size without reducers or bushings. All fittings between the boiler and valves shall be of steel.

311 When the maximum allowable working pressure exceeds 125 lb. per sq. in., the bottom blow-off pipe shall have two valves, or a valve and a cock, and such valves, or valve and cock, shall be extra heavy, except that on a boiler having multiple blow-off pipes, a single master valve may be placed on the common blow-off pipe from the boiler, in which case only one valve on each individual blow-off is required.

312 A bottom blow-off pipe when exposed to direct furnace heat shall be protected by fire-brick, a substantial cast-iron removable sleeve or a covering of non-conducting material.

313 An opening in the boiler setting for a blow-off pipe shall be arranged to provide for free expansion and contraction.

314 *Feed Piping.* The feed pipe of a boiler shall have an open end or ends. Wherever globe valves are used on feed piping, the inlet shall be under the disc of the valve.

315 The feedwater shall discharge at about three-fifths the length of a horizontal return tubular boiler from the front head (except a horizontal return tubular boiler equipped with an auxiliary feedwater heating and circulating device), above the central rows of tubes, when the diameter of the boiler exceeds 36 in. The feed pipe shall be carried through the head or shell near the front end with a brass or steel boiler bushing, and securely fastened inside the shell above the tubes.

316 Feedwater shall not discharge in a boiler close to riveted joints in the shell or to furnace sheets.

317 The feed pipe shall be provided with a check valve near the boiler and a valve or cock between the check valve and the boiler, and when two or more boilers are fed from a common source, there shall also be a globe valve on the branch to each boiler, between the check valve and the source of supply.

318 When a pump, inspirator or injector is required to supply feedwater to a boiler plant of over 50 h. p., more than one such appliance shall be provided.

319 *Lamphrey Fronts.* Each boiler fitted with a Lamphrey boiler furnace mouth protector, or similar appliance, having valves on the pipes connecting them to the boiler, shall have these valves locked or sealed *open.* Such valves when used, shall be of the straight-way type.

320 *Water Column Pipes.* The minimum size of pipes connecting the water column to a boiler shall be 1 in. Water-glass fittings or gage cocks may be connected direct to the boiler.

321 The water connections to the water column of a boiler shall be of brass and shall be provided with a cross to facilitate cleaning. Either the water column or this connection shall be fitted with a drain cock or drain valve with a suitable connection to the ashpit, or other safe point of waste. The water column blow-off pipe shall be at least ¾ in.

322 The steam connection to the water column of a horizontal return tubular boiler shall be taken from the top of the shell or the upper part of the head; the water connection shall be taken from a point not less than 6 in. below the center line of the shell.

Setting

323 *Methods of Support.* A horizontal return tubular boiler over 78-in. in diameter shall be supported from steel lugs by the outside suspension type of setting, independent of the boiler side walls. The lugs shall be so designed that the load is properly distributed between the rivets attaching them to the shell and so that not more than two of these rivets come in the same longitudinal line on each lug. The distance girthwise of the boiler from the centers of the bottom rivets to the centers of the top rivets attaching the lugs shall be not less than 12 in. The other rivets used shall be spaced evenly between these points. If more than four lugs are used they shall be set in four pairs.

324 A horizontal return tubular boiler over 54 in., and up to and including 78 in. in diameter, shall be supported by the outside suspension type of setting, or at four points by not less than eight steel or cast-iron brackets set in pairs. A horizontal return tubular boiler up to and including 54 in. in diameter shall be supported by the outside suspension type of setting, or by not less than two steel or cast-iron brackets on each side.

325 Lugs or brackets, when used to support boilers, shall be properly fitted to the surfaces to which they are attached. The shearing stress on the rivets used for attaching the lugs or brackets shall not exceed 8 per cent of the strength given in Par. 16.

326 Wet-bottom stationary boilers shall have a space of not less than 12 in. between the bottom of the boiler and the floor line, with access for inspection.

327 *Access and Firing Doors.* The minimum size of an access door to be placed in a boiler setting shall be 12 × 16 in. or equivalent area, 11 in. to be the least dimension in any case.

328 A water tube boiler which is fired by hand shall have firing door or doors of the inward opening type unless such doors are provided with substantial latching devices to prevent them from being blown open by pressure on the furnace side.

Hydrostatic Tests

329 *Hydrostatic Pressure Tests.* After a boiler has been completed, it shall be subjected to a hydrostatic test of one and one-half times the maximum allowable working pressure. The pressure shall be under proper control so that in no case shall the required test pressure be exceeded by more than 6 per cent.

330 During a hydrostatic test, the safety valve or valves shall be removed or each valve disc shall be held to its seat by means of a testing clamp and not by screwing down the compression screw upon the spring.

Stamping

331 *Stamping of Boilers.* In laying out shell plates, furnace sheets and heads in the boiler shop, care shall be taken to leave at least one of the stamps, specified in Par. 36 of these Rules, so located as to be plainly visible when the boiler is completed; except that the tube sheets of a vertical fire-tube boiler and butt straps shall have at least a portion of such stamps visible sufficient for identification when the boiler is completed.

332 Each boiler shall conform in every detail to these Rules, and shall be distinctly stamped with the symbol as shown in Fig. 19, denoting that the boiler was constructed in accordance therewith. Each boiler shall also be stamped by the builder with a serial number and

with the builder's name either in full or abbreviated, as indicated in Fig. 20. The height of the letters and figures used in stamping shall be not less than ¼ in. and this stamp shall be placed directly below or alongside The American Society of Mechanical Engineers' stamp.

Fig. 19 Official Symbol for Stamp to Denote The American Society of Mechanical Engineers Uniform Standard

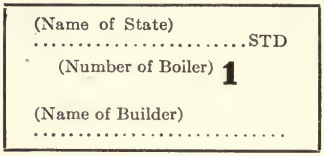

Fig. 20 Form of Stamp Proposed for the Boiler Manufacturer

333 *Location of Stamps.* The location of stamps shall be as follows:

a On horizontal return tubular boilers—on the front head, above the central rows of tubes.

b On horizontal flue boilers—on the front head, above the flues.

c On traction, portable or stationary boilers of the locomotive type or Star water-tube boilers—on the furnace end, above the handhole.

d On vertical fire tube and vertical submerged tube boilers—on the shell above the fire door.

e On water-tube boilers, Babcock & Wilcox, Stirling, Heine and Robb-Mumford standard types—on a head above the manhole opening, preferably on the flanging of the manhole opening.

f On vertical boilers, Climax or Hazleton type—on the top head.

g On Cahall or Wickes vertical water tube boilers—on the upper drum, above the manhole opening.

h On Scotch marine boilers—on the front head, above the center or right-hand furnace.

i On Economic boilers—on the front head, above the central row of tubes.

j For other types and new designs—in a conspicuous location.

334 The American Society of Mechanical Engineers' standard stamp and the boiler builder's stamps shall not be covered by insulating or other material.

PART I—SECTION II

BOILERS USED EXCLUSIVELY FOR LOW PRESSURE STEAM AND HOT WATER HEATING AND HOT WATER SUPPLY

(THIS DOES NOT APPLY TO ECONOMIZERS OR FEED WATER HEATERS.)

Boiler Materials

335 The Rules for power boilers shall apply:

a To all steel plate *hot-water* boilers over 60 in. in diameter.

b To all steel plate *hot-water* boilers where the grate area exceeds 10 sq. ft. and the maximum allowable working pressure exceeds 50 lb. per sq. in.

c Under other conditions, the following rules shall apply.

336 Specifications are given in these Rules, Pars. 23 to 178, for the important materials used in the construction of boilers, and where given, the materials shall conform thereto.

337 Flange steel may be used entirely for the construction of steam heating boilers covered in this section, but in no case shall steel of less than ¼ in. in thickness, nor tube sheets or heads of less than 5/16 in. in thickness be used.

Maximum Allowable Working Pressure

338 The maximum allowable working pressure shall not exceed 15 lb. per sq. in. on a boiler built under these Rules to be used exclusively for low pressure steam heating.

339 A boiler to be used exclusively for low-pressure steam heating, may be constructed of cast-iron, or of cast-iron excepting connecting nipples and bolts, or wholly of steel or wrought-iron, or of steel and partially cast-iron, or of steel or wrought-iron with cast-iron mud rings, door frames and manhole flanges.

340 All steel plate, *hot-water* and *steam-heating* boilers shall have a factor of safety of not less than 5.

Boiler Joints

341 Longitudinal lap joints will be allowed on boilers to be used exclusively for low pressure *steam* heating, when the maximum allowable working pressure does not exceed 15 lb. per sq. in., and the diameter of the boiler shell does not exceed 60 in.

342 The longitudinal joints of a horizontal return tubular boiler if of the lap type, shall be not over 12 ft. in length.

343 In a *hot-water* boiler to be used exclusively for heating buildings or hot water supply when the diameter does not exceed 60 in. and the grate area does not exceed 10 sq. ft., longitudinal lap joints will be allowed.

When the grate area exceeds 10 sq. ft. and the diameter of the boiler does not exceed 60 in. longitudinal lap joints will be allowed providing the maximum allowable working pressure does not exceed 50 lb. per sq. in.

344 *Protection of Joints.* When a boiler is built wholly or partially of steel and is used exclusively for low pressure *steam* heating, or when a *hot-water* boiler is used exclusively for heating buildings or for hot-water supply, it shall not be necessary to water jacket the rivets in the fire-box where one end of each rivet is exposed to the fire or direct radiant heat from the fire, provided any one of the following conditions is fulfilled :

a Where the ends of the rivets away from the fire are protected by means of natural drafts of cold air induced in the regular operation of the boiler ;

b Where the ends of the rivets away from the fire are in the open air ;

c Where the rivets are protected by the usual charges of fresh fuel, which is not burned in contact with the rivets.

Washout Holes

345 A boiler used for hot-water supply shall be provided with washout holes for the removal of any sediment that may accumulate therein.

Boiler Openings

346 *Flanged Connections.* Openings in boilers having flanged connections shall have the flanges conform to the American Standard

given in Tables 15 or 16 of the Appendix, for the corresponding pipe size, and shall have the corresponding drilling for bolts or studs.

Safety Valves

347 *Outlet Connections for Safety and Water Relief Valves.* Every boiler shall have proper outlet connections for the required safety, or water relief valve or valves, independent of any other connection outside of the boiler or any internal pipe in the boiler, the area of the opening to be at least equal to the aggregate area of all of the safety valves with which it connects. A screwed connection may be used for attaching a safety valve to a heating boiler. This rule applies to all sizes of safety valves.

348 *Safety Valves.* Each *steam* boiler shall be provided with one or more safety valves of the spring-pop type which cannot be adjusted to a higher pressure than 15 lb. per sq. in.

349 *Water Relief Valves.* Each *hot-water* boiler shall be provided with one or more water relief valves with open discharges having outlets in plain sight.

350 A *hot-water* boiler built for a maximum allowable working pressure of 30 lb. per sq. in. and used exclusively for heating buildings, or for hot-water supply, shall be provided with a water relief valve or valves, which cannot be adjusted for a pressure in excess of 30 lb. per sq. in.

351 No safety or water relief valve shall be smaller than 1 in. nor greater than 4½ in. nominal size.

352 When two or more safety or water relief valves are used on a boiler they may be single or twin valves.

353 Safety or water relief valves shall be connected to boilers independent of other connections and be attached directly or as close as possible to the boiler, without any intervening pipe or fittings, except the Y-base forming a part of the twin valve or the shortest possible connection. A safety or water relief valve shall not be connected to an internal pipe in the boiler. Safety valves shall be connected so as to stand upright, with the spindle vertical, when possible.

354 No shut-off of any description shall be placed between the safety or water relief valves and boilers, nor on discharge pipes between them and the atmosphere.

355 When a discharge pipe is used its area shall be not less than the area of the valve or aggregate area of the valves with which it

connects, and the discharge pipe shall be fitted with an open drain to prevent water from lodging in the upper part of the valve or in the pipe. When an elbow is placed on a safety or water relief valve discharge pipe, it shall be located close to the valve outlet or the pipe shall be securely anchored and supported. The safety or water relief valves shall be so located and piped that there will be no danger of scalding attendants.

356 Each safety valve used on a *steam* heating boiler shall have a substantial lifting device which shall be so connected to the disc that the latter can be lifted from its seat a distance of not less than

TABLE 9 ALLOWABLE SIZES OF SAFETY VALVES FOR HEATING BOILERS

Water Evaporated per Sq. Ft. of Grate Surface per Hr., Lb.		75	100	160	160	200	240
Maximum allowable Working Pressure, Lb. per Sq. In.		Zero to 25 Lb.	Over 25 to 50 Lb.	Over 50 to 100 Lb.	Over 100 to 150 Lb.	Over 150 to 200 Lb.	Over 200 Lb.
Diameter of Valve, In.	Area of Valve, Sq. In.	Area of Grate, Sq. Ft.					
1	0.7854	2.00	2.50	2.75	3.25	3.5	3.75
1¼	1.2272	3.25	4.00	4.25	5.00	5.5	5.75
1½	1.7671	4.50	5.50	6.00	7.25	8.0	8.50
2	3.1416	8.00	9.75	10.75	13.00	14.0	15.00
2½	4.9087	12.50	15.00	16.50	20.00	22.0	23.00
3	7.0686	17.75	21.50	24.00	29.00	31.5	33.25
3½	9.6211	24.00	29.50	32.50	39.50	43.0	45.25
4	12.5660	31.50	38.25	42.50	51.50	56.0	59.00
4½	15.9040	40.00	48.50	53.50	65.00	71.0	74.25

one-tenth of the nominal diameter of the seat when there is no pressure on the boiler. A relief valve used on a hot-water heating boiler need not have a lifting device.

357 Every safety valve or water relief valve shall have plainly stamped on the body or cast thereon the manufacturer's name or trade mark and the pressure at which it is set to blow. The seats and discs of safety or water relief valves shall be made of non-ferrous material.

358 The minimum size of safety or water relief valve or valves for each boiler shall be governed by the grate area of the boiler, as shown by Table 9.

When the conditions exceed those on which Table 9 is based, the following formula for bevel and flat seated valves shall be used:

$$A = \frac{W \times 70}{P} \times 11$$

in which

A = area of direct spring-loaded safety valve per square foot of grate surface, sq. in.

W = weight of water evaporated per square foot of grate surface per second, lb.

P = pressure (absolute) at which the safety valve is set to blow, lb. per sq. in.

359 *Double Grate Down Draft Boilers.* In determining the number and size of safety valves or water relief valves the grate area shall equal the area of the upper grate plus one-half of the area of the lower grate.

360 *Boilers Fired With Oil or Gas.* In determining the number and size of safety or water relief valve or valves for a boiler using gas or liquid fuel, 15 sq. ft. of heating surface shall be equivalent to one square foot of grate area. If the size of grate for use of coal is evident from the boiler design, such size may be the basis for the determination of the safety valve capacity.

Steam and Water Gages

361 *Steam Gages.* Each *steam* boiler shall have a steam gage connected to the steam space or to the water column or its steam connection. The steam gage shall be connected to a syphon or equivalent device of sufficient capacity to keep the gage tube filled with water and so arranged that the gage cannot be shut off from the boiler except by a cock placed near the gage and provided with a tee or lever handle arranged to be parallel to the pipe in which it is located when the cock is open. Connections to gages shall be of brass, copper or bronze composition. The dial of a steam gage for a *steam* heating boiler shall be graduated to not less than 30 lb.

362 *Pressure or Altitude Gages.* Each *hot-water* boiler shall have a gage connected in such a manner that it cannot be shut off from the boiler except by a cock with tee or lever handle, placed on the pipe near the gage. The handle of the cock shall be parallel to the pipe in which it is located when the cock is open. Connections to gages shall be made of brass, copper or bronze composition. The dial of

the pressure or altitude gage shall be graduated to not less than 1½ times the maximum allowable working pressure.

363 *Thermometers.* Each *hot-water* boiler shall have a thermometer so located and connected that it shall be easily readable when observing the water pressure or altitude. The thermometer shall be so located that it shall at all times indicate the temperature in deg. fahr., of the water in the boiler.

Fittings and Appliances

364 *Bottom Blow-off Pipes.* Each boiler shall have a blow-off pipe, fitted with a valve or cock, in direct connection with the lowest water space practicable.

365 *Damper Regulators.* When a pressure damper regulator is used, the boiler pressure pipe shall be connected to the steam space of the boiler.

366 *Water Glasses.* Each *steam* boiler shall have one or more water glasses.

367 *Gage Cocks.* Each *steam* boiler shall have two or more gage cocks located within the range of the visible length of the water glass.

368 *Water Column Pipes.* The minimum size of pipes connecting the water column of a boiler shall be 1 in. Water-glass fittings or gage cocks may be connected direct to the boiler. The steam connection to the water column of a horizontal return tubular boiler shall be taken from the top of shell or the upper part of the head; the water connection shall be taken from a point not less than 6 in. below the center line of the shell. No connections, except for damper regulator, drains or steam gages, shall be placed on the pipes connecting a water column to a boiler.

Methods of Setting

369 Wet-bottom steel plate boilers shall have a space of not less than 12 in. between the bottom of the boiler and the floor line with access for inspection.

370 *Access Doors.* The minimum size of access door used in boiler settings shall be 12 × 16 in. or equivalent area, the least dimension being 11 in.

371 The longitudinal joints of a horizontal return tubular boiler shall be located above the fire-line.

Hydrostatic Tests

372 A shop test of 60 lb. per sq. in. hydrostatic pressure shall be applied to steel or cast-iron boilers or to the sections of cast-iron boilers which are used exclusively for low pressure *steam* heating.

373 *Hot-water* boilers for a maximum allowable working pressure not exceeding 30 lb. per sq. in. used exclusively for heating buildings or for hot-water supply, when constructed of cast-iron, or of cast-iron excepting the connecting nipples and bolts, shall be subjected to a shop test of 60 lb. per sq. in. hydrostatic pressure applied to the boiler or the sections thereof.

374 A maximum allowable working pressure in excess of 30 lb. per sq. in. will be allowed on a *hot-water* boiler constructed of cast-iron, or of cast-iron excepting the connecting nipples and bolts, used exclusively for heating buildings or for hot-water supply, provided such boilers or their sections have been subjected to a shop hydrostatic test of *two and one-half times* the actual working pressure.

375 Individual shop inspection shall be required only for boilers which come under the rules for power boilers.

Stamping

376 Each plate of a completed boiler shall show a sufficient portion of the plate maker's stamp for identification.

377 *Name.* All boilers referred to in this section shall be plainly and permanently marked with the manufacturer's name and the maximum allowable working pressure.

PART II—EXISTING INSTALLATIONS

Maximum Allowable Working Pressure

378 The maximum allowable working pressure on the shell of a boiler or drum shall be determined by the strength of the weakest course, computed from the thickness of the plate, the tensile strength of the plate, the efficiency of the longitudinal joint, the inside diameter of the course and the factor of safety allowed by these Rules.

$\frac{TS \times t \times E}{R \times FS}$ = maximum allowable working pressure, lb. per sq. in.

where

TS = ultimate tensile strength of shell plates, lb. per sq. in.

t = thickness of shell plate, in weakest course, in.

E = efficiency of longitudinal joint, method of determining which is given in Par. 181, of these Rules

R = inside radius of the weakest course of the shell or drum, in.

FS = factor of safety allowed by these Rules

379 Boilers in service one year after these Rules become effective shall be operated with a factor of safety of at least 4 by the formula, Par. 378. Five years after these Rules become effective, the factor of safety shall be at least 4.5. In no case shall the maximum allowable working pressure on old boilers be increased, unless they are being operated at a lesser pressure than would be allowable for new boilers, in which case the changed pressure shall not exceed that allowable for new boilers of the same construction.

380 The age limit of a horizontal return tubular boiler having a longitudinal lap joint and carrying over 50 lb. pressure shall be 20 years, except that no lap joint boiler shall be discontinued from service solely on account of age until 5 years after these Rules become effective.

381 Second-hand boilers, by which are meant boilers where both the ownership and location are changed, shall have a factor of safety of at least 5½, by the formula Par. 378, one year after these Rules become effective, unless constructed in accordance with the Rules contained in Part I, when the factor shall be at least 5.

382 *Cast-Iron Headers and Mud Drums.* The maximum allowable working pressure on a water tube boiler, the tubes of which are secured to cast-iron or malleable iron headers, or which have cast-iron mud drums, shall not exceed 160 lb. per sq. in.

383 *Steam Heating Boilers.* The maximum allowable working pressure shall not exceed 15 lb. per sq: in. on a boiler used exclusively for low pressure *steam* heating.

384 No pressure shall be allowed on a boiler on which a crack is discovered along the longitudinal riveted joint.

Strength of Materials

385 *Tensile Strength.* When the tensile strength of steel or wrought-iron shell plates is *not* known, it shall be taken as 55,000 lb. per sq. in. for steel and 45,000 lb. for wrought-iron.

386 *Strength of Rivets in Shear.* In computing the ultimate strength of rivets in shear the following values in pounds per square inch of the cross-sectional area of the rivet shank shall be used:

Iron rivets in single shear	38,000
Iron rivets in double shear	76,000
Steel rivets in single shear	44,000
Steel rivets in double shear	88,000

The cross-sectional area shall be that of the rivet shank after driving.

387 *Crushing Strength of Mild Steel.* The resistance to crushing of mild steel shall be taken at 95,000 lb. per sq. in. of cross-sectional area.

TABLE 10 SIZES OF RIVETS BASED ON PLATE THICKNESS

Thickness of plate	¼″	5/16″	3/8″	7/16″	½″	9/16″
Diameter of rivet after driving	11/16″	11/16″	¾″	¾″	13/16″	13/16″
Thickness of plate	5/8″	11/16″	¾″	13/16″	⅞″	—
Diameter of rivet after driving	15/16″	15/16″	15/16″	1 1/16″	1 1/16″	—

388 *Rivets.* When the diameter of the rivet holes in the longitudinal joints of a boiler is *not* known, the diameter and cross-sectional area of rivets, after driving may be ascertained from Table 10, or by cutting out one rivet in the body of the joint.

Safety Valves for Power Boilers

389 The safety valve capacity of each boiler shall be such that the safety valve or valves will discharge all the steam that can be generated by the boiler without allowing the pressure to rise more than 6 per cent above the maximum allowable working pressure, or more than 6 per cent above the highest pressure to which any valve is set.

390 One or more safety valves on every boiler shall be set at or below the maximum allowable working pressure. The remaining valves may be set within a range of 3 per cent above the maximum allowable working pressure, but the range of setting of all of the valves on a boiler shall not exceed 10 per cent of the highest pressure to which any valve is set.

391 Safety valve capacity may be checked in any one of three different ways, and if found sufficient, additional capacity need not be provided:

a By making an accumulation test, by shutting off all other steam discharge outlets from the boiler and forcing the fires to the maximum. The safety valve equipment shall be sufficient to prevent an excess pressure beyond 6 per cent as specified in Par. 389.

b By measuring the maximum amount of fuel that can be burned and computing the corresponding evaporative capacity upon the basis of the heating value of the fuel. See Appendix, Pars. 421 to 427.

c By determining the maximum evaporative capacity by measuring the feedwater. The sum of the safety valve capacities shall be equal to or greater than the maximum evaporative capacity of the boiler.

392 In case either of the methods outlined in sections *b* or *c* of Par. 391 is employed, the safety valve capacities shall be taken at the maximum values given in Table 8 for spring loaded pop safety valves, or 0.66 times the maximum values given in Table 8, for lever safety valves.

393 When additional valve capacity is required, any valves added shall conform to the requirements in Part I of these Rules.

394 No valve of any description shall be placed between the safety valve and the boiler, nor on the discharge pipe between the safety valve and the atmosphere. When a discharge pipe is used, it

shall be not less than the full size of the valve, and the discharge pipe shall be fitted with an open drain to prevent water lodging in the upper part of the safety valve or in the pipe. If a muffler is used on a safety valve it shall have sufficient outlet area to prevent back pressure from interfering with the proper operation and discharge capacity of the valve. The muffler plates or other devices shall be so constructed as to avoid any possibility of restriction of the steam passages due to deposit. When an elbow is placed on a safety valve discharge pipe, it shall be located close to the safety valve outlet or the pipe shall be securely anchored and supported. All safety valve discharges shall be so located or piped as to be carried clear from running boards or working platforms used in controlling the main stop valves of boilers or steam headers.

Fittings and Appliances

395 *Water Glasses and Gage Cocks.* Each steam boiler shall have at least one water glass, the lowest visible part of which shall be not less than 2 in. above the lowest permissible water level.

396 Each boiler shall have three or more gage cocks, located within the range of the visible length of the water glass, when the maximum allowable working pressure exceeds 15 lb. per sq. in., except when such boiler has two water glasses with independent connections to the boiler, located on the same horizontal line and not less than 2 ft. apart.

397 No outlet connections, except for damper regulator, feedwater regulator, drains or steam gages, shall be placed on the pipes connecting a water column to a power boiler.

398 *Steam Gages.* Each steam boiler shall have a steam gage connected to the steam space or to the water column or to its steam connection. The steam gage shall be connected to a syphon or equivalent device of sufficient capacity to keep the gage tube filled with water and so arranged that the gage cannot be shut off from the boiler except by a cock placed near the gage and provided with a tee or lever handle arranged to be parallel to the pipe in which it is located when the cock is open. Connections to gages shall be of brass, copper or bronze composition.

399 *Stop Valves.* Each steam outlet from a power boiler (except safety valve connections) shall be fitted with a stop valve located as close as practicable to the boiler.

400 When a stop valve is so located that water can accumulate, ample drains shall be provided.

401 *Bottom Blow-Off Pipes.* Each boiler shall have a blow-off pipe fitted with a valve or cock, in direct connection with the lowest water space practicable.

402 When the maximum allowable working pressure exceeds 125 lb. per sq. in., the blow-off pipe shall be extra heavy from boiler to valve or valves, and shall run full size without reducers or bushings. All fittings between the boiler and valve shall be steel, extra heavy malleable iron or extra heavy cast-iron.

403 When the maximum allowable working pressure exceeds 125 lb. per sq. in., each bottom blow-off pipe shall be fitted with an extra heavy valve or cock. Preferably two (2) valves, or a valve and a cock should be used on each blow-off in which case such valves, or valve and cock, shall be extra heavy.

404 A bottom blow-off pipe when exposed to direct furnace heat, shall be protected from the products of combustion by fire-brick, a substantial cast-iron removable sleeve, or a covering of non-conducting material.

405 An opening in the boiler setting for a blow-off pipe shall be arranged to provide for free expansion and contraction.

406 *Feed Piping.* The feed pipe of a steam boiler operated at more than 15 lb. per sq. in. maximum allowable working pressure, shall be provided with a check valve near the boiler and a valve or cock between the check valve and the boiler, and when two or more boilers are fed from a common source, there shall also be a globe valve on the branch to each boiler, between the check valve and the source of supply. When a globe valve is used on a feed pipe, the inlet shall be under the disc of the valve.

407 *Lamphrey Fronts.* Each boiler fitted with a Lamphrey boiler furnace mouth protector, or similar appliance, having valves on the pipes connecting them to the boiler, shall have these valves locked or sealed *open*. Such valves, when used, shall be of the straightway type.

Hydrostatic Pressure Tests

408 *Test Pressure.* When a hydrostatic test is applied the required test pressure shall be one and one-half times the maximum allowable working pressure. The pressure shall be under proper control so that in no case shall the required test pressure be exceeded by more than 2 per cent.

409 During a hydrostatic test of a boiler, the safety valve or valves shall be removed or each valve disc shall be held to its seat by means of a testing clamp and not by screwing down the compression screw upon the spring.

APPENDIX

Efficiency of Joints

410 *Efficiency of Riveted Joints.* The ratio which the strength of a unit length of a riveted joint has to the same unit length of the solid plate is known as the efficiency of the joint and shall be calculated by the general method illustrated in the following examples:

TS = tensile strength stamped on plate, lb. per sq. in.
t = thickness of plate, in.
b = thickness of butt strap, in.
P = pitch of rivets, in., on row having greatest pitch
d = diameter of rivet after driving, in. = diameter of rivet hole

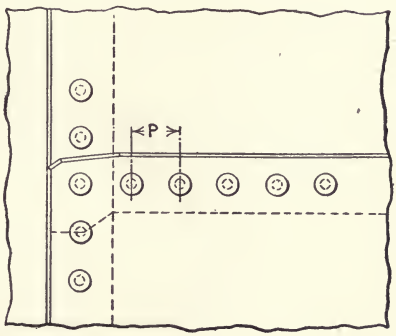

Fig. 21 Example of Lap Joint, Longitudinal or Circumferential, Single-Riveted

a = cross-sectional area of rivet after driving, sq. in.
s = shearing strength of rivet in single shear, lb. per sq. in., as given in Par. 16
S = shearing strength of rivet in double shear, lb. per sq. in., as given in Par. 16
c = crushing strength of mild steel, lb. per sq. in., as given in Par. 15
n = number of rivets in single shear in a unit length of joint
N = number of rivets in double shear in a unit length of joint.

411 *Example:* Lap joint, longitudinal or circumferential, single-riveted.

A = strength of solid plate = $P \times t \times TS$
B = strength of plate beween rivet holes = $(P—d)t \times TS$
C = shearing strength of one rivet in single shear = $n \times s \times a$
D = crushing strength of plate in front of one rivet = $d \times t \times c$
Divide B, C or D (whichever is the least) by A, and the quotient will be the efficiency of a single-riveted lap joint as shown in Fig. 21.

TS = 55,000 lb. per sq. in.
t = ¼ in. = 0.25 in.
P = 1⅝ in. = 1.625 in.
d = $\frac{11}{16}$ in. = 0.6875 in.
a = 0.3712 sq. in.
s = 44,000 lb. per sq. in.
c = 95,000 lb. per sq. in.
A = 1.625 × 0.25 × 55,000 = 22,343
B = (1.625—0.6875) 0.25 × 55,000 = 12,890
C = 1 × 44,000 × 0.3712 = 16,332
D = 0.6875 × 0.25 × 95,000 = 16,328

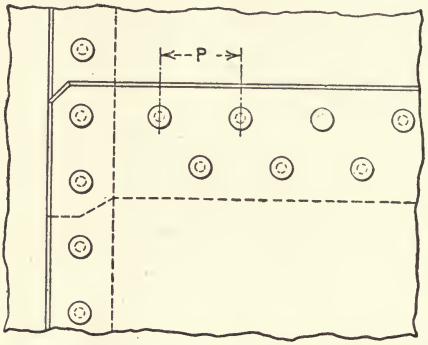

FIG. 22 EXAMPLE OF LAP JOINT, LONGITUDINAL OR CIRCUMFERENTIAL, DOUBLE-RIVETED

$$\frac{12{,}890\ (B)}{22{,}343\ (A)} = 0.576 = \text{efficiency of joint}$$

412 *Example:* Lap joint, longitudinal or circumferential, double-riveted.

A = strength of solid plate = $P \times t \times TS$
B = strength of plate between rivet holes = $(P—d)\ t \times TS$
C = shearing strength of two rivets in single shear = $n \times s \times a$
D = crushing strength of plate in front of two rivets = $n \times d \times t \times c$
Divide B, C or D (whichever is the least) by A, and the quotient will be the efficiency of a double-riveted lap joint, as shown in Fig. 22.

TS = 55,000 lb. per sq. in.
t = $\frac{5}{16}$ in. = 0.3125 in.
P = 2⅞ in. = 2.875 in.
d = ¾ in. = 0.75 in.
a = 0.4418 sq. in.
s = 44,000 lb. per sq. in.
c = 95,000 lb. per sq. in.
A = 2.875 × 0.3125 × 55,000 = 49,414
B = (2.875—0.75) 0.3125 × 55,000 = 36,523
C = 2 × 44,000 × 0.4418 = 38,878
D = 2 × 0.75 × 0.3125 × 95,000 = 44,531

$$\frac{36{,}523\ (B)}{49{,}414\ (A)} = 0.739 = \text{efficiency of joint}$$

413 *Example:* Butt and double strap joint, double-riveted.

A = strength of solid plate = $P \times t \times TS$

B = strength of plate between rivet holes in the outer row = $(P—d)\ t \times TS$

C = shearing strength of two rivets in double shear, plus the shearing strength of one rivet in single shear = $N \times S \times a + n \times s \times a$

D = strength of plate between rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row = $(P—2d)\ t \times TS + n \times s \times a$

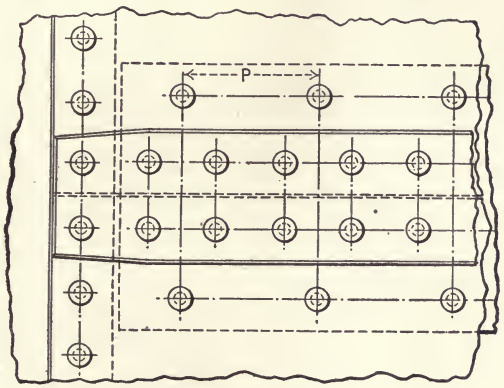

FIG. 23 EXAMPLE OF BUTT AND DOUBLE STRAP JOINT, DOUBLE-RIVETED

E = strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one rivet in the outer row = $(P—2d)\ t \times TS + d \times b \times c$

F = crushing strength of plate in front of two rivets, plus the crushing strength of butt strap in front of one rivet = $N \times d \times t \times c + n \times d \times b \times c$

G = crushing strength of plate in front of two rivets, plus the shearing strength of one rivet in single shear = $N \times d \times t \times c + n \times s \times a$

H = strength of butt straps between rivet holes in the inner row = $(P—2d)\ 2b \times TS$. This method of failure is not possible for thicknesses of butt straps required by these Rules and the computation need only be made for old boilers in which thin butt straps have been used. For this reason this method of failure will not be considered in other joints.

Divide B, C, D, E, F, G or H (whichever is the least) by A, and the quotient will

be the efficiency of a butt and double strap joint, double-riveted, as shown in Fig. 23.

$TS = 55{,}000$ lb. per sq. in.
$t = \frac{3}{8}$ in. $= 0.375$ in.
$b = \frac{5}{16}$ in. $= 0.3125$ in.
$P = 4\frac{7}{8}$ in. $= 4.875$ in.
$d = \frac{7}{8}$ in. $= 0.875$ in.
$a = 0.6013$ sq. in.
$s = 44{,}000$ lb. per sq. in.
$S = 88{,}000$ lb. per sq. in.
$c = 95{,}000$ lb. per sq. in.

Number of rivets in single shear in a unit length of joint $= 1$.
Number of rivets in double shear in a unit length of joint $= 2$.

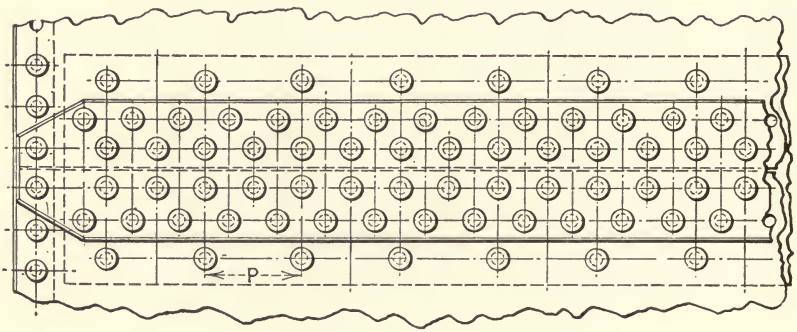

FIG. 24 EXAMPLE OF BUTT AND DOUBLE STRAP JOINT, TRIPLE-RIVETED

$A = 4.875 \times 0.375 \times 55{,}000 = 100{,}547$
$B = (4.875 - 0.875)\ 0.375 \times 55{,}000 = 82{,}500$
$C = 2 \times 88{,}000 \times 0.6013 + 1 \times 44{,}000 \times 0.6013 = 132{,}286$
$D = (4.875 - 2 \times 0.875)\ 0.375 \times 55{,}000 + 1 \times 44{,}000 \times 0.6013 = 90{,}910$
$E = (4.875 - 2 \times 0.875)\ 0.375 \times 55{,}000 + 0.875 \times 0.3125 \times 95{,}000 = 90{,}429$
$F = 2 \times 0.875 \times 0.375 \times 95{,}000 + 0.875 \times 0.3125 \times 95{,}000 = 88{,}320$
$G = 2 \times 0.875 \times 0.375 \times 95{,}000 + 1 \times 44{,}000 \times 0.6013 = 88{,}800$

$$\frac{82{,}500\ (B)}{100{,}547\ (A)} = 0.820 = \text{efficiency of joint}$$

414 *Example:* Butt and double strap joint, triple-riveted.

A = strength of solid plate $= P \times t \times TS$

B = strength of plate between rivet holes in the outer row $= (P - d)\ t \times TS$

C = shearing strength of four rivets in double shear, plus the shearing strength of one rivet in single shear $= N \times S \times a + n \times s \times a$

D = strength of plate between rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row $= (P - 2d)\ t \times TS + n \times s \times a$

E = strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one rivet in the outer row = $(P—2d)\ t \times TS + d \times b \times c$

F = crushing strength of plate in front of four rivets, plus the crushing strength of butt strap in front of one rivet = $N \times d \times t \times c + n \times d \times b \times c$

G = crushing strength of plate in front of four rivets, plus the shearing strength of one rivet in single shear = $N \times d \times t \times c + n \times s \times a$

Divide B, C, D, E, F or G (whichever is the least) by A, and the quotient will be the efficiency of a butt and double strap joint, triple-riveted, as shown in Fig. 24.

TS = 55,000 lb. per sq. in.
t = $\frac{3}{8}$ in. = 0.375 in.
b = $\frac{5}{16}$ in. = 0.3125 in.
P = 6½ in. = 6.5 in.
d = $\frac{13}{16}$ in. = 0.8125 in.
a = 0.5185 sq. in.
s = 44,000 lb. per sq. in.
S = 88,000 lb. per sq. in.
c = 95,000 lb. per sq. in.

Number of rivets in single shear in a unit length of joint = 1.

Number of rivets in double shear in a unit length of joint = 4.

$A = 6.5 \times 0.375 \times 55{,}000 = 134{,}062$
$B = (6.5—0.8125)\ 0.375 \times 55{,}000 = 117{,}304$
$C = 4 \times 88{,}000 \times 0.5185 + 1 \times 44{,}000 \times 0.5185 = 205{,}326$
$D = (6.5—2 \times 0.8125)\ 0.375 \times 55{,}000 + 1 \times 44{,}000 \times 0.5185 = 123{,}360$
$E = (6.5—2 \times 0.8125)\ 0.375 \times 55{,}000 + 0.8125 \times 0.3125 \times 95{,}000 = 124{,}667$
$F = 4 \times 0.8125 \times 0.375 \times 95{,}000 + 1 \times 0.8125 \times 0.3125 \times 95{,}000 = 139{,}902$
$G = 4 \times 0.8125 \times 0.375 \times 95{,}000 + 1 \times 44{,}000 \times 0.5185 = 138{,}595$

$$\frac{117{,}304\ (B)}{134{,}062\ (A)} = 0.875 = \text{efficiency of joint}$$

415 *Example:* Butt and double strap joint, quadruple-riveted.

A = strength of solid plate = $P \times t \times TS$

B = strength of plate between rivet holes in the outer row = $(P—d)\ t \times TS$

C = shearing strength of eight rivets in double shear, plus the shearing strength of three rivets in single shear = $N \times S \times a + n \times s \times a$

D = strength of plate between rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row = $(P—2d)\ t \times TS + 1 \times s \times a$

E = strength of plate between rivet holes in the third row, plus the shearing strength of two rivets in the second row in single shear and one rivet in single shear in the outer row = $(P—4d)\ t \times TS + n \times s \times a$

F = strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one rivet in the outer row = $(P—2d)\ t \times TS + d \times b \times c$

G = strength of plate between rivet holes in the third row, plus the crushing strength of butt strap in front of two rivets in the second row and one rivet in the outer row = $(P-4d)\ t\times TS+n\times d\times b\times c$

H = crushing strength of plate in front of eight rivets, plus the crushing strength of butt strap in front of three rivets = $N\times d\times t\times c+n\times d\times b\times c$

I = crushing strength of plate in front of eight rivets, plus the shearing strength of two rivets in the second row and one rivet in the outer row, in single shear = $N\times d\times t\times c+n\times s\times a$

Divide B, C, D, E, F, G, H or I (whichever is the least) by A, and the quotient will be the efficiency of a butt and double strap joint quadruple-riveted, as shown in Fig. 25.

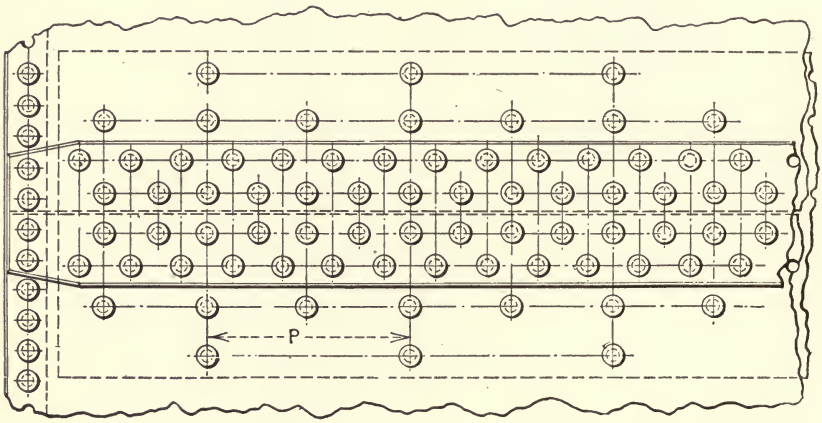

FIG. 25 EXAMPLE OF BUTT AND DOUBLE STRAP JOINT, QUADRUPLE-RIVETED

TS = 55,000 lb. per sq. in.
t = ½ in. = 0.5 in.
b = $\frac{7}{16}$ in. = 0.4375 in.
P = 15 in.
d = $\frac{15}{16}$ in. = 0.9375 in.
a = 0.6903 sq. in.
s = 44,000 lb. per sq. in.
S = 88,000 lb. per sq. in.
c = 95,000 lb. per sq. in.

Number of rivets in single shear in a unit length of joint = 3.
Number of rivets in double shear in a unit length of joint = 8.

$A = 15\times 0.5\times 55{,}000 = 412{,}500$
$B = (15-0.9375)\ 0.5\times 55{,}000 = 386{,}718$
$C = 8\times 88{,}000\times 0.6903+3\times 44{,}000\times 0.6903 = 577{,}090$
$D = (15-2\times 0.9375)\ 0.5\times 55{,}000+1\times 44{,}000\times 0.6903 = 391{,}310$
$E = (15-4\times 0.9375)\ 0.5\times 55{,}000+3\times 44{,}000\times 0.6903 = 400{,}494$
$F = (15-2\times 0.9375)\ 0.5\times 55{,}000+0.9375\times 0.4375\times 95{,}000 = 399{,}902$
$G = (15-4\times 0.9375)\ 0.5\times 55{,}000+3\times 0.9375\times 0.4375\times 95{,}000 = 426{,}269$
$H = 8\times 0.9375\times 0.5\times 95{,}000+3\times 0.9375\times 0.4375\times 95{,}000 = 473{,}145$
$I = 8\times 0.9375\times 0.5\times 95{,}000+3\times 44{,}000\times 0.6903 = 447{,}369$

$$\frac{386{,}718\ (B)}{412{,}500\ (A)} = 0.937 = \text{efficiency of joint}$$

416 *Example:* Butt and double strap joint, quintuple-riveted.

A = strength of solid plate = $P \times t \times TS$

B = strength of plate between rivet holes in the outer row = $(P-d)\ t \times TS$

C = shearing strength of 16 rivets in double shear, plus the shearing strength of seven rivets in single shear = $N \times S \times a + n \times s \times a$

D = strength of plate between rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row = $(P-2d)\ t \times TS + 1 \times s \times a$

E = strength of plate between rivet holes in the third row, plus the shearing strength of two rivets in the second row in single shear and one rivet in single shear in the outer row = $(P-4d)\ t \times TS + 3 \times s \times a$

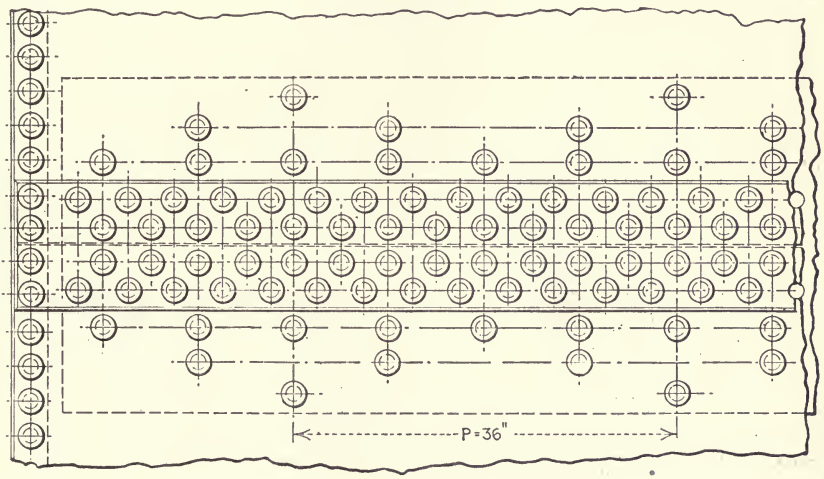

FIG. 26 EXAMPLE OF BUTT AND DOUBLE STRAP JOINT, QUINTUPLE-RIVETED

F = strength of plate between rivet holes in the fourth row, plus the shearing strength of four rivets in the third row, two rivets in the second row and one rivet in the outer row in single shear = $(P-8d)\ t \times TS + n \times s \times a$

G = strength of plate between rivet holes in the second row, plus the crushing strength of butt strap in front of one rivet in the outer row = $(P-2d)\ t \times TS + d \times b \times c$

H = strength of plate between rivet holes in the third row, plus the crushing strength of butt strap in front of two rivets in the second row and one rivet in the outer row = $(P-4d)\ t \times TS + 3 \times d \times b \times c$

I = strength of plate between rivet holes in the fourth row, plus the crushing strength of butt strap in front of four rivets in the third row, two rivets in the second row and one rivet in the outer row = $(P-6d)\ t \times TS + n \times d \times b \times c$

J = crushing strength of plate in front of 16 rivets, plus the crushing strength of butt strap in front of seven rivets $= N \times d \times t \times c + n \times d \times b \times c$

K = crushing strength of plate in front of 16 rivets, plus the shearing strength of four rivets in the third row, two rivets in the second row and one rivet in the outer row in single shear $= N \times d \times t \times c + n \times s \times a$

Divide B, C, D, E, F, G, H, I, J or K (whichever is the least) by A, and the quotient will be the efficiency of a butt and double strap joint, quintuple-riveted, as shown in Fig. 26 or Fig. 27.

$TS = 55{,}000$ lb. per sq. in.
$t = \frac{3}{4}$ in. $= 0.75$ in.
$b = \frac{1}{2}$ in. $= 0.5$ in.
$P = 36$ in.
$d = 1\frac{5}{16}$ in. $= 1.3125$ in.
$a = 1.3529$ sq. in.
$s = 44{,}000$ lb. per sq. in.
$S = 88{,}000$ lb. per sq. in.
$c = 95{,}000$ lb. per sq. in.

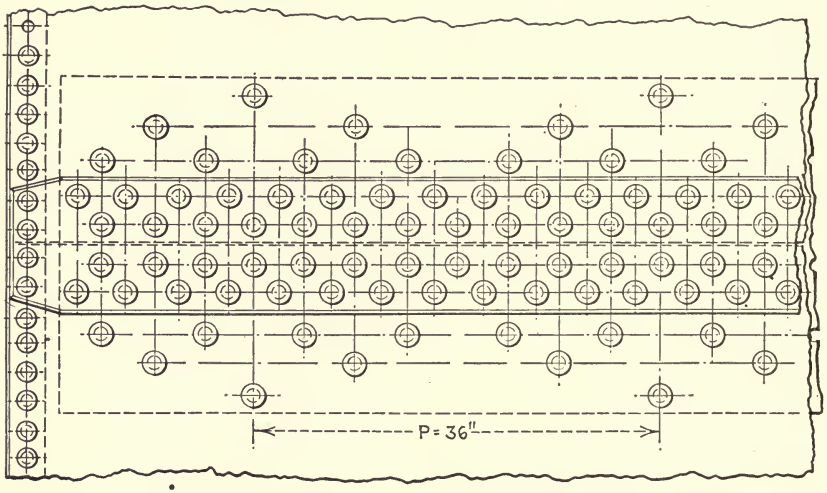

FIG. 27 EXAMPLE OF BUTT AND DOUBLE STRAP JOINT, QUINTUPLE-RIVETED

Number of rivets in single shear in a unit length of joint = 7.
Number of rivets in double shear in a unit length of joint = 16.

$A = 36 \times 0.75 \times 55{,}000 = 1{,}485{,}000$
$B = (36 - 1.3125)\ 0.75 \times 55{,}000 = 1{,}430{,}860$
$C = 16 \times 88{,}000 \times 1.3529 + 7 \times 44{,}000 \times 1.3529 = 2{,}321{,}576$
$D = (36 - 2 \times 1.3125)\ 0.75 \times 55{,}000 + 1 \times 44{,}000 \times 1.3529 = 1{,}436{,}246$
$E = (36 - 4 \times 1.3125)\ 0.75 \times 55{,}000 + 3 \times 44{,}000 \times 1.3529 = 1{,}447{,}020$
$F = (36 - 8 \times 1.3125)\ 0.75 \times 55{,}000 + 7 \times 44{,}000 \times 1.3529 = 1{,}468{,}568$
$G = (36 - 2 \times 1.3125)\ 0.75 \times 55{,}000 + 1.3125 \times 0.5 \times 95{,}000 = 1{,}439{,}064$
$H = (36 - 4 \times 1.3125)\ 0.75 \times 55{,}000 + 3 \times 1.3125 \times 0.5 \times 95{,}000 = 1{,}455{,}472$
$I = (36 - 8 \times 1.3125)\ 0.75 \times 55{,}000 + 7 \times 1.3125 \times 0.5 \times 95{,}000 = 1{,}488{,}141$
$J = 16 \times 1.3125 \times 0.75 \times 95{,}000 + 7 \times 1.3125 \times 0.5 \times 95{,}000 = 1{,}932{,}266$
$K = 16 \times 1.3125 \times 0.75 \times 95{,}000 + 7 \times 44{,}000 \times 1.3529 = 1{,}912{,}943$

$$\frac{1{,}430{,}860\ (B)}{1{,}485{,}000\ (A)} = 0.963 = \text{efficiency of joint}$$

417 Figs. 28 and 29 illustrate other joints that may be used. The butt and double strap joint with straps of equal width shown in Fig. 28 may be so designed that it will have an efficiency of from 82 to 84 per cent and the saw-tooth joint shown in Fig. 29 so that it will have an efficiency of from 92 to 94 per cent.

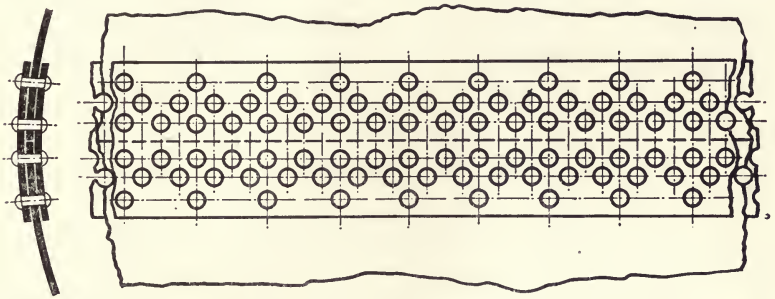

FIG. 28 ILLUSTRATION OF BUTT AND DOUBLE STRAP JOINT WITH STRAPS OF EQUAL WIDTH

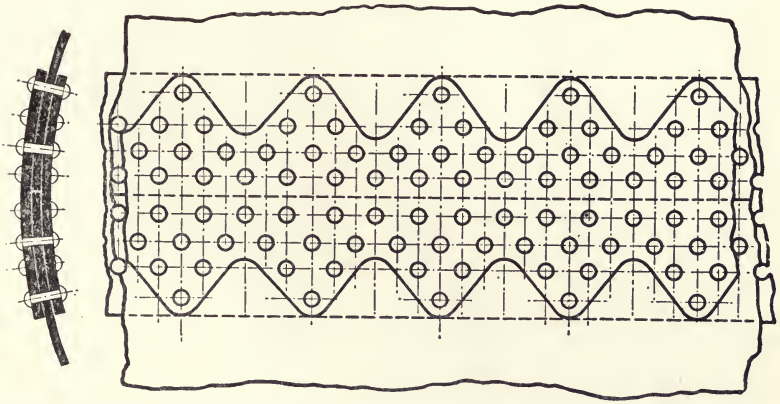

FIG. 29 ILLUSTRATION OF BUTT AND DOUBLE STRAP JOINT OF THE SAW-TOOTH TYPE

Braced and Stayed Surfaces

418 The allowable loads based on the net cross-sectional areas of staybolts with V-threads, are computed from the following formulae. The use of Whitworth threads with other pitches is permissible.

The formula for the diameter of a staybolt at the bottom of a V-thread is:

$$D - (P \times 1.732) = d$$

where

D = diameter of staybolt over the threads, in.
P = pitch of threads, in.
d = diameter of staybolt at bottom of threads, in.
1.732 = a constant

When U. S. threads are used, the formula becomes

$$D - (P \times 1.732 \times 0.75) = d$$

Tables 11 and 12 give the allowable loads on net cross-sectional areas for staybolts with V-threads, having 12 and 10 threads per inch.

TABLE 11. ALLOWABLE LOADS ON STAYBOLTS WITH V-THREADS, 12 THREADS PER INCH

Outside Diameter of Staybolts, In.		Diameter at Bottom of Thread, In.	Net Cross-Sectional Area (at Bottom of Thread), Sq. In.	Allowable Load at 7500 Lb. Stress, per Sq. In.
3/4	0.7500	0.6057	0.288	2160
13/16	0.8125	0.6682	0.351	2632
7/8	0.8750	0.7307	0.419	3142
15/16	0.9375	0.7932	0.494	3705
1	1.0000	0.8557	0.575	4312
1 1/16	1.0625	0.9182	0.662	4965
1 1/8	1.1250	0.9807	0.755	5662
1 3/16	1.1875	1.0432	0.855	6412
1 1/4	1.2500	1.1057	0.960	7200
1 5/16	1.3125	1.1682	1.072	8040
1 3/8	1.3750	1.2307	1.190	8925
1 7/16	1.4375	1.2932	1.313	9849
1 1/2	1.5000	1.3557	1.444	10830

TABLE 12. ALLOWABLE LOADS ON STAYBOLTS WITH V-THREADS, 10 THREADS PER INCH

Outside Diameter of Staybolts, In.		Diameter at Bottom of Thread, In.	Net Cross-Sectional Area (at Bottom of Thread), Sq. In.	Allowable Load at 7500 Lb. Stress per Sq. In.
1¼	1.2500	1.0768	0.911	6832
1 5/16	1.3125	1.1393	1.019	7642
1⅜	1.3750	1.2018	1.134	8505
1 7/16	1.4375	1.2643	1.255	9412
1½	1.5000	1.3268	1.382	10365
1 9/16	1.5625	1.3893	1.515	11362
1⅝	1.6250	1.4518	1.655	12412

419 Table 13 shows the allowable loads on net cross-sectional areas of round stays or braces.

TABLE 13. ALLOWABLE LOADS ON ROUND BRACES OR STAY RODS

Minimum Diameter of Circular Stay, In.		Net Cross-sectional Area of Stay, in Sq. In.	Allowable Stress, in Lb. per Sq. In., Net Cross-sectional Area		
			6000	8500	9500
			Allowable Load, in Lb., on Net Cross-sectional Area		
1	1.0000	0.7854	4712	6676	7462
1 1/16	1.0625	0.8866	5320	7536	8423
1⅛	1.1250	0.9940	5964	8449	9443
1 3/16	1.1875	1.1075	6645	9414	10521
1¼	1.2500	1.2272	7363	10431	11658
1 5/16	1.3125	1.3530	8118	11501	12854
1⅜	1.3750	1.4849	8909	12622	14107
1 7/16	1.4375	1.6230	9738	13796	15419
1½	1.5000	1.7671	10603	15020	16787
1 9/16	1.5625	1.9175	11505	16298	18216
1⅝	1.6250	2.0739	12443	17628	19702
1 11/16	1.6875	2.2365	13419	19010	21247
1¾	1.7500	2.4053	14432	20445	22852
1 13/16	1.8125	2.5802	15481	21932	24512
1⅞	1.8750	2.7612	16567	23470	26231
1 15/16	1.9375	2.9483	17690	25061	28009
2	2.0000	3.1416	18850	26704	29845
2⅛	2.1250	3.5466	21280	30147	33693
2¼	2.2500	3.9761	23857	33797	37773
2⅜	2.3750	4.4301	26580	37656	42086
2½	2.5000	4.9087	29452	41724	46632
2⅝	2.6250	5.4119	32471	46001	51413
2¾	2.7500	5.9396	35638	50487	56426
2⅞	2.8750	6.4918	38951	55181	61673
3	3.0000	7.0686	42412	60083	67152

420 Table 14 gives the net areas of segments of heads for use in computing stays.

TABLE 14. NET AREAS OF SEGMENTS OF HEADS

Height from Tubes to Shell, In.	Diameter of Boiler, In.												
	24	30	36	42	48	54	60	66	72	78	84	90	96
	Area to be stayed, Sq. In.												
8	28	33	37	40	43	47	51	53	55	58	60	63	65
8½	35	41	46	51	55	59	63	66	70	74	76	80	82
9	42	49	56	62	67	72	76	82	86	90	92	95	98
9½	50	58	66	70	80	86	91	96	101	105	111	116	119
10	57	68	77	85	93	99	106	112	117	123	129	132	137
10½	66	78	89	98	107	114	123	131	135	142	147	153	160
11	74	88	100	111	121	130	138	147	155	161	169	174	183
11½	83	99	112	124	137	146	156	165	173	181	189	196	204
12	91	109	125	139	151	163	174	184	194	203	213	219	230
12½	...	120	138	153	167	180	193	204	216	224	234	243	252
13	....	132	151	168	183	197	211	224	235	247	256	267	279
13½	....	143	164	183	200	216	230	246	258	270	282	293	302
14	....	155	178	199	217	234	250	266	280	294	305	319	331
14½	....	167	192	215	235	254	271	287	303	318	333	345	360
15	....	178	206	231	252	273	291	309	326	343	357	372	386
15½	....	.....	220	247	271	291	312	332	350	368	382	400	417
16	....	.....	235	263	289	312	334	355	374	394	411	423	443
16½	....	.....	249	281	308	332	357	380	399	420	436	457	475
17	....	.....	264	297	326	353	378	402	425	447	467	486	502
17½	....	.....	.....	314	345	374	400	426	449	471	494	516	536
18	....	.....	.....	331	365	396	424	450	476	500	520	543	564
18½	....	.....	.....	349	384	417	448	476	501	526	552	577	598
19	....	.....	.....	366	404	439	470	500	529	555	580	604	631
19½	....	.....	.....	384	424	461	496	528	558	584	613	641	663
20	....	.....	.....	401	444	483	519	552	583	613	642	667	699
20½	....	.....	.....	.....	464	505	543	578	613	643	675	706	729
21	....	.....	.....	.....	485	528	568	604	640	673	705	733	766
21½	....	.....	.....	.....	505	551	594	632	669	703	739	766	797
22	....	.....	.....	.....	526	574	618	658	697	734	769	800	835
22½	....	.....	.....	.....	.....	597	643	687	726	765	800	835	867
23	....	.....	.....	.....	.....	620	668	713	754	796	830	869	906
23½	....	.....	.....	.....	.....	642	695	740	784	827	866	904	945
24	....	.....	.....	.....	.....	667	719	768	814	859	897	939	978
24½	....	.....	.....	.....	.....	689	745	797	843	892	934	975	1018
25	....	.....	.....	.....	.....	714	771	825	875	922	966	1010	1051
25½	....	.....	.....	.....	.....	737	798	855	907	956	1003	1047	1092
26	....	.....	.....	.....	.....	761	824	882	936	987	1035	1083	1126
26½	....	.....	.....	.....	.....	.....	850	909	968	1024	1073	1120	1167
27	....	.....	.....	.....	.....	.....	877	939	998	1053	1106	1157	1202
27½	....	.....	.....	.....	.....	.....	904	968	1030	1089	1145	1195	1243
28	....	.....	.....	.....	.....	.....	930	997	1060	1120	1177	1232	1279
28½	....	.....	.....	.....	.....	.....	.....	1028	1092	1157	1211	1270	1321
29	....	.....	.....	.....	.....	.....	.....	1056	1123	1187	1248	1305	1360
29½	....	.....	.....	.....	.....	.....	.....	1084	1155	1221	1284	1347	1400
30	....	.....	.....	.....	.....	.....	.....	1115	1187	1255	1321	1382	1442
30½	....	.....	.....	.....	.....	.....	.....	.....	1218	1290	1358	1424	1480
31	....	.....	.....	.....	.....	.....	.....	.....	1252	1324	1394	1459	1523
31½	....	.....	.....	.....	.....	.....	.....	.....	1286	1359	1433	1496	1561
32	....	.....	.....	.....	.....	.....	.....	.....	1317	1394	1467	1538	1605
32½	....	.....	.....	.....	.....	.....	.....	.....	.....	1430	1508	1575	1650
33	....	.....	.....	.....	.....	.....	.....	.....	.....	1465	1542	1617	1687
33½	....	.....	.....	.....	.....	.....	.....	.....	.....	1500	1578	1655	1733
34	....	.....	.....	.....	.....	.....	.....	.....	.....	1536	1617	1695	1770
34½	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1654	1735	1816
35	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1692	1775	1856
35½	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1810	1900
36	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1857	1941
36½	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1984
37	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	2026

Safety Valves

421 *Method of Computing Table* 8. The discharge capacity of a safety valve is expressed in equations 2 and 3 as the product of C and H. The discharge capacities are given in Table 8 for each valve size at the pressures shown and are calculated for various valve sizes, pressures and for three different lifts. The discharge capacities are proportional to the lifts, so that intermediate values may be obtained from the Table by interpolation.

C = total weight or volume of fuel of any kind burned per hour at time of maximum forcing, lb. or cu. ft.

H = the heat of combustion, B.t.u. per lb. or cu. ft. of fuel used.

D = diameter of valve seat, in.

L = vertical lift of valve disc, in., measured immediately after the sudden lift due to the pop.

P = absolute boiler pressure or gage pressure plus 14.7 lb. per sq. in.

1100 = the number of B.t.u. required to change a pound of feed water at 100 deg. fahr. into a pound of steam.

The boiler efficiency is assumed as 75 per cent.

The coefficient of discharge, in Napier's formula, is taken as 96 per cent.

$$\frac{C\times H\times 0.75}{1100\times 3600}=\frac{3.1416\times D\times L\times 0.707\times P\times 0.96}{70}$$ for valve with 45-deg. seat. (1)

$CH = 160{,}856\times P\times D\times L$ for valve with bevel seat at 45 deg. (2)

$CH = 227{,}487\times P\times D\times L$ for valve with flat seat at 90 deg. (3)

Method of Checking the Safety Valve Capacity by Measuring the Maximum Amount of Fuel that can be Burned

422 The maximum weight of fuel that can be burned is determined by a test. The weight of steam generated per hour is found from the formula:

$$W=\frac{C\times H\times 0.75}{1100}$$ where

W = weight of steam generated per hour, lb.

C = total weight of fuel burned per hour at time of maximum forcing, lb.

H = the heat of combustion of the fuel, B.t.u. per lb. (see Par. 427).

The sum of the safety valve capacities marked on the valves as provided for in the Rules shall be equal to or greater than the maximum evaporative capacity of the boiler.

Table 8 may be used for determining the number of safety valves required as illustrated in the following examples:

423 *Example 1:* A boiler at the time of maximum forcing uses 2150 lb. of Illinois coal per hour of 12,100 B.t.u. per lb. Boiler pressure, 225 lb. per sq. in. gage.

$$2150 \times 12{,}100 = CH = 26{,}015{,}000$$

Table 8 shows that two 3½-in. bevel seated valves with 0.11 in. lift, or one 3-in. bevel seated valve with 0.10 in. lift and one 3½-in. bevel seated valve with 0.11 in. lift, would discharge the steam generated.

424 *Example 2:* Wood shavings of heat of combustion of 6400 B.t.u. per lb. are burned under a boiler at the maximum rate of 2000 lb. per hour. Boiler pressure, 100 lb. per sq. in. gage.

$$2000 \times 6400 = CH = 12{,}800{,}000$$

Table 8 shows that two 3½-in. bevel seated valves with 0.11 in. lift, or one 3-in. bevel seated valve with 0.08 in. lift and one 4-in. bevel seated valve with 0.12 in. lift, would discharge the steam generated.

425 *Example 3:* An oil-fired boiler at maximum forcing uses 1000 lb. of crude oil (Texas) per hour. Boiler pressure, 275 lb. per sq. in. gage.

$$1000 \times 18{,}500 = CH = 18{,}500{,}000$$

Table 8 shows that two 3½-in. bevel seated valves with 0.06 in. lift, or two 3-in. flat seated valves with 0.05 in. lift, or two 2½-in. flat seated valves with 0.06 in. lift, would discharge the steam generated.

426 *Example 4:* A boiler fired with natural gas consumes 3000 cu. ft. per hour. The working pressure is 150 lb. per sq. in. gage.

$$3000 \times 960 = CH = 2{,}880{,}000$$

Table 8 shows that two 1¼-in. bevel seated valves with 0.05 in. lift, or two 1-in. flat seated valves with 0.04 in. lift, would discharge the steam generated.

427 For the purpose of checking the safety valve capacity as described in Par. 422, the following values of heats of combustion of various fuels in B.t.u. per lb. or per cu. ft. may be used:

	B. t. u. per lb.
Semi-bituminous coal	14,500
Anthracite	13,700
Screenings	12,500
Coke	13,500
Wood, hard or soft, kiln dried	7,700
Wood, hard or soft, air dried	6,200
Wood shavings	6,400
Peat, air dried, 25 per cent moisture	7,500
Lignite	10,000
Kerosene	20,000
Petroleum, crude oil, Penn	20,700
Petroleum, crude oil, Texas	18,500

	B. t. u. per cu. ft.
Natural gas	960
Blast furnace gas	100
Producer gas	150
Water gas, uncarburetted	290

TABLE 15

AMERICAN STANDARD 125-LB WORKING PRESSURE PER SQ. IN. STANDARD FLANGE FITTINGS, STRAIGHT SIZES (SEE FIG. 30)

	Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15
A-A	Face to face	7	7½	8	9	10	11	12	13	14	15	16	17	18	20	22	24	28	29
A	Center to face	3½	3¾	4	4½	5	5½	6	6½	7	7½	8	8½	9	10	11	12	14	14½
B	Center to face of long radius ells	5	5½	6	6½	7	7¾	8½	9	9½	10¼	11½	12¾	14	15¼	16½	19	21½	22¾
C	Center to face of 45-deg. ells	1¾	2	2¼	2½	3	3	3½	4	4	4½	5	5½	5½	6	6½	7½	7½	8
D	Face to face laterals	7½	8	9	10½	12	13	14½	15	15½	17	18	20½	22	24	25½	30	33	34½
E	Center to face	5¾	6¼	7	8	9½	10	11½	12	12½	13½	14½	16½	17½	19½	20½	24½	27	28½
F	Center to face	1¾	1¾	2	2½	2½	3	3	3	3	3½	3½	4	4½	4½	5	5½	6	6
G	Face to face reducer						6	6½	7	7½	8	9	10	11	11½	12	14	16	17
	Diameter of flange	4	4½	5	6	7	7½	8½	9	9¼	10	11	12½	13½	15	16	19	21	22¼
	Thickness of flange	7/16	1/2	9/16	5/8	11/16	3/4	13/16		15/16	1	1 1/16	1 1/8	1 3/16	1 1/8	1 3/16	1¼	1⅜	1⅜
	Diameter of bolt circle	3	3⅜	3⅞	4¾	5½	6	7	7½	7¾	8½	9½	10¾	11¾	13¼	14¼	17	18¾	20
	No. of bolts	4	4	4	4	4	4	4	8	8	8	8	8	8	12	12	12	12	16
	Diameter of bolts	7/16	7/16	1/2	5/8	5/8	5/8	5/8	5/8	3/4	3/4	3/4	3/4	3/4	3/4	7/8	7/8	1	1
	Minimum metal thickness of body	7/16	7/16	7/16	7/16	7/16	7/16	7/16	1/2	1/2	1/2	5/8	5/8	11/16	3/4	13/16	7/8	7/8	7/8

	Size	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
A-A	Face to face	30	33	36	40	44	46	48	50	52	54	56	58	60	62	64	66	68
A	Center to face	15	16½	18	20	22	23	24	25	26	27	28	29	30	31	32	33	34
B	Center to face of long radius ells	24	26½	29	31½	34	36½	39	41½	44	46½	49	51½	54	56½	59	61½	64
C	Center to face of 45-deg. ells	8	8½	9½	10	11	13	14	15	16	17	18	19	20	21	22	23	24
D	Face to face laterals	36½	39	43	46	49½	53	56	59									
E	Center to face	30	32	35	37½	40½	44	46½	49									
F	Center to face	6½	7	8	8½	9	9	9½	10									
G	Face to face reducer	18	19	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
	Diameter of flange	23½	25	27½	29½	32	34¼	36½	38¾	41¾	43¾	46	48¾	50¾	53	55¼	57¼	59½
	Thickness of flange	1 1/16	1 1/16	1 1/16	1 1/16	1⅞	2	2 1/16	2⅛	2¼	2 5/16	2⅜	2⅜	2½	2⅝	2⅝	2 11/16	2¾
	Diameter of bolt circle	21¼	22¾	25	27¼	29½	31¾	34	36	38½	40½	42¾	45¼	47¼	49½	51¾	53¾	56
	No. of bolts	16	16	20	20	20	24	28	28	28	32	32	32	36	36	40	40	44
	Diameter of bolts	1	1⅛	1⅛	1¼	1¼	1¼	1¼	1⅜	1⅜	1½	1½	1½	1⅝	1⅝	1⅝	1⅝	1⅝
	Minimum metal thickness of body	1	1 1/16	1⅛	1 3/16	1¼	1 5/16	1⅜	1 7/16	1½	1 9/16	1⅝	1 11/16	1¾	1 13/16	1⅞	1 15/16	2

NOTES:—Figures given are for center to face and for face to face finished dimensions. Where necessary manufacturers will make suitable allowances in patterns before casting.

Laterals do not extend beyond the 30-in. size at the present time. Box wrench to be used on bolting for large sizes.

Square head bolts with hexagonal nuts are recommended. 1⅝ in. diameter and larger stud with a nut at each end is satisfactory.

Hexagonal nuts for pipe sizes 1 in. to 46 in. can be conveniently pulled up with open wrenches of minimum design of heads. Hexagonal nuts for pipe sizes 48 in. to 100 in. can be conveniently pulled up with socket wrenches.

Flanges to be spot bored for nuts for sizes 32 in. to 100 in. inclusive.

TABLE 16

AMERICAN STANDARD, 250-LB. WORKING PRESSURE PER SQ. IN., EXTRA HEAVY FLANGE FITTINGS, STRAIGHT SIZES (SEE FIG. 30)

	Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15
A-A	Face to face	8	8½	9	10	11	12	13	14	15	16	17	18	20	21	23	26	30	31
A	Center to face	4	4¼	4½	5	5½	6	6½	7	7½	8	8½	9	10	10½	11½	13	15	15½
B	Center to face of long radius ells	5	5½	6	6½	7	7¾	8½	9	9½	10¼	11½	12¾	14	15¼	16½	19	21½	22¾
C	Center to face of 45-deg. ells	2	2½	2¾	3	3½	3½	4	4½	4½	5	5½	6	6	6½	7	8	8½	9
D	Face to face, laterals	8½	9½	11	11½	13	14	15½	16½	18	18½	21½	23½	25½	27½	29½	33½	37½	39½
E	Center to face, laterals	6½	7¼	8½	9	10½	11	12½	13½	14½	15	17½	19	20½	22½	24	27½	31	33
F	Center to face, laterals	2	2¼	2½	2½	2½	3	3	3	3½	3½	4	4½	5	5	5½	6	6½	6½
G	Face to face, reducer	....	....	....	....	....	6	6½	7	7½	8	9	10	11	11½	12	14	16	17
	Diameter of flange	4½	5	6	6½	7½	8¼	9	10	10½	11	12½	14	15	16¼	17½	20½	23	24½
	Thickness of flange	11/16	3/4	13/16	7/8	1	1 1/8	1 3/16	1¼	1 5/16	1 3/8	1 7/16	1½	1 5/8	1¾	1 7/8	2	2 1/8	2 3/16
	Diameter of bolt circle	3¼	3¾	4½	5	5⅞	6⅝	7¼	7⅞	8½	9¼	10⅝	11⅞	13	14	15¼	17¾	20¼	21½
	Number of bolts	4	4	4	4	4	8	8	8	8	8	12	12	12	12	16	16	20	20
	Diameter of bolts	½	½	5/8	5/8	¾	¾	¾	¾	¾	¾	¾	⅞	⅞	1	1	1⅛	1⅛	1¼
	Minimum metal thickness of body	½	½	½	½	9/16	9/16	9/16	5/8	5/8	11/16	¾	13/16	15/16	1	1 1/16	1	1⅛	1 1/16

	Size	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
A-A	Face to face	33	36	39	41	45	48	52	55	58	61	65	68	71	74	78	81	84
A	Center to face	16½	18	19½	20½	22½	24	26	27½	29	30½	32½	34	35½	37	39	40½	42
B	Center to face of long radius ells	24	26½	29	31½	34	36½	39	41½	44	46½	49	51½	54	56½	59	61½	64
C	Center to face of 45-deg. ells	9½	10	10½	11	12	13	14	15	16	17	18	19	20	21	22	23	24
D	Face to face, laterals	42	45½	49	53	57½	....	....	....	....	....	....	....	....	....	....	....	....
E	Center to face, laterals	34½	37½	40½	43½	47½	....	....	....	....	....	....	....	....	....	....	....	....
F	Center to face, laterals	7½	8	8½	9½	10	....	....	....	....	....	....	....	....	....	....	....	....
G	Face to face, reducer	18	19	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
	Diameter of flange	25½	28	30½	33	36	38¼	40¾	43	45¼	47½	50	52¼	54½	57	59¼	61½	63
	Thickness of flange	2 1/4	2 3/8	2½	2 5/8	2¾	2 13/16	2 15/16	3	3 1/8	3 1/4	3 3/8	3 7/16	3 5/8	3 11/16	3¾	3 7/8	4
	Diameter of bolt circle	22½	24¾	27	29¼	32	34½	37	39¼	41½	43½	46	48	50¼	52¾	55	57¼	60¾
	Number of bolts	20	24	24	24	24	28	28	28	28	28	32	32	36	36	36	40	40
	Diameter of bolts	1¼	1¼	1⅜	1½	1⅝	1⅝	1⅝	1¾	1⅞	1⅞	1⅞	1⅞	1⅞	1⅞	2	2	2
	Minimum metal thickness of body	1¼	1⅜	1½	1 9/16	1⅝	1 13/16	1⅞	2	2⅛	2¼	2⅜	2 7/16	2 9/16	2 11/16	2 13/16	2⅞	3

NOTES:—Figures given are for center to face and for face to face finished dimensions. Where necessary manufacturers will make suitable allowances in patterns before casting.

Laterals do not extend beyond the 24 in. size at the present time. Box wrench to be used on bolting for large sizes.

Square head bolts with hexagonal nuts are recommended. 1⅝ in. diameter and larger stud with a nut at each end is satisfactory.

Hexagonal nuts for pipe sizes 1 in. to 16 in. can be conveniently pulled up with wrenches of minimum design of heads. Hexagonal nuts for pipe sizes 18 in. to 48 in. can be conveniently pulled up with socket wrenches.

Distance between inside edges of bolt holes and raised face to be ⅛ in.

Flanges to be spot bored for nuts.

Thickness of flanges given in table includes raised face.

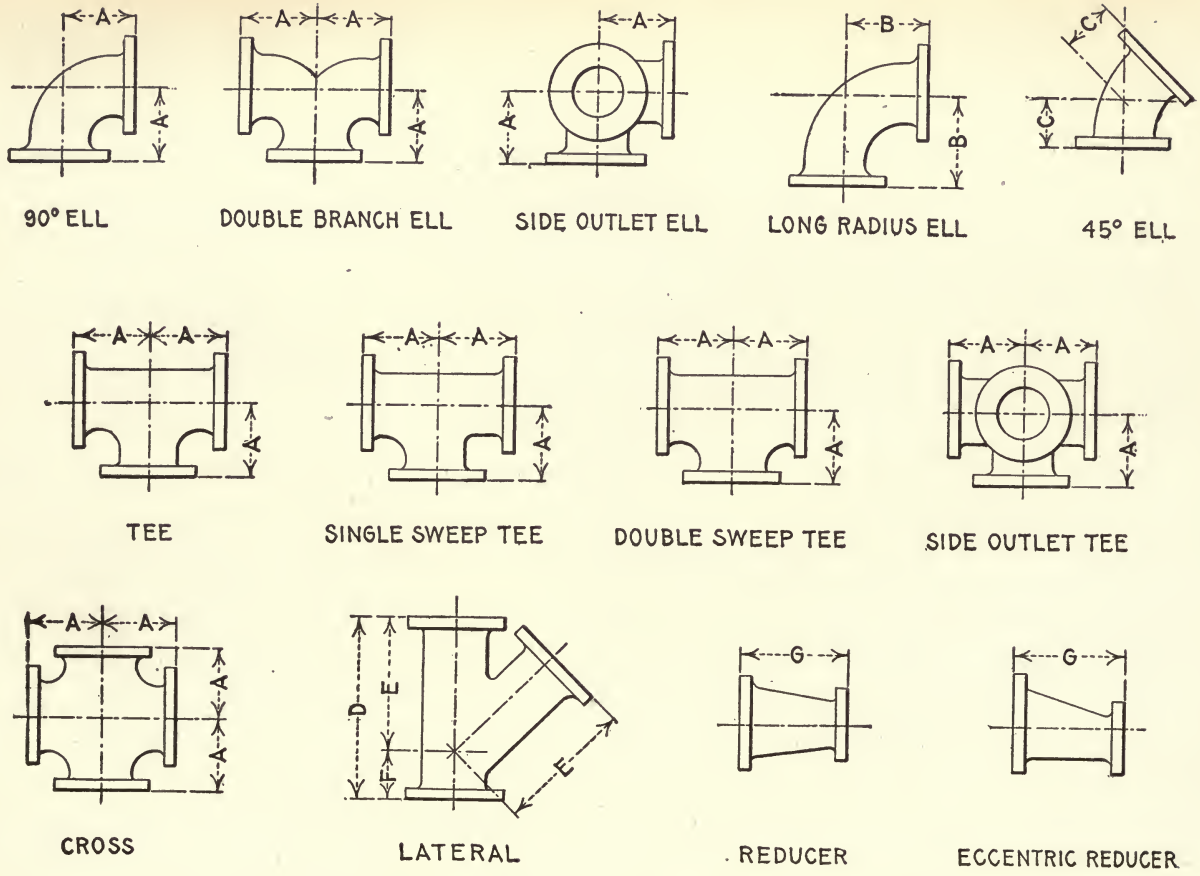

FIG. 30 STANDARD TYPES OF FLANGE FITTINGS DIMENSIONED IN TABLES 15 AND 16

Fusible Plugs

428 Fusible plugs, if used, shall be filled with tin with a melting point between 400 and 500 deg. fahr.

429 The least diameter of fusible metal shall be not less than ½ in., except for maximum allowable working pressures of over 175 lb. per sq. in. or when it is necessary to place a fusible plug in a tube, in which case the least diameter of fusible metal shall be not less than ⅜ in.

430 Each boiler may have one or more fusible plugs, located as follows:

a In Horizontal Return Tubular Boilers—in the rear head, not less than 2 in. above the upper row of tubes, the measurement to be taken from the line of the upper surface of tubes to the center of the plug, and projecting through the sheet not less than 1 in.

b In Horizontal Flue Boilers—in the rear head, on a line with the highest part of the boiler exposed to the products of combustion, and projecting through the sheet not less than 1 in.

c In Traction, Portable or Stationary Boilers of the Locomotive Type or Star Water Tube Boilers—in the highest part of the crown sheet, and projecting through the sheet not less than 1 in.

d In Vertical Fire-tube Boilers—in an outside tube, not less than one-third the length of the tube above the lower tube sheet.

e In Vertical Fire-tube Boilers, Corliss Type—in a tube, not less than one-third the length of the tube above the lower tube sheet.

f In Vertical Submerged Tube Boilers—in the upper tube sheet, and projecting through the sheet not less than 1 in.

g In Water-tube Boilers, Horizontal Drums, Babcock & Wilcox Type—in the upper drum, not less than 6 in. above the bottom of the drum, over the first pass of the products of combustion, and projecting through the sheet not less than 1 in.

h In Stirling Boilers, Standard Type—in the front side of the middle drum, not less than 4 in. above the bottom of the drum, and projecting through the sheet not less than 1 in.

i In Stirling Boilers, Superheater Type—in the front drum, not less than 6 in. above the bottom of the drum, exposed to the products of combustion, and projecting through the sheet not less than 1 in.

j In Water-tube Boilers, Heine Type—in the front course of the drum, not less than 6 in. above the bottom of the drum, and projecting through the sheet not less than 1 in.

k In Robb-Mumford Boilers, Standard Type—in the bottom of the steam and water drum, 24 in. from the center of the rear neck, and projecting through the sheet not less than 1 in.

l In Water-tube Boilers, Almy Type—in a tube or fitting exposed to the products of combustion.

m In Vertical Boilers, Climax or Hazelton Type—in a tube or center drum not less than one-half the height of the shell, measuring from the lowest circumferential seam.

n In Cahall Vertical Water-tube Boilers—in the inner sheet of the top drum, not less than 6 in. above the upper tube sheet, and projecting through the sheet not less than 1 in.

o In Wickes Vertical Water-tube Boilers—in the shell of the top drum and not less than 6 in. above the upper tube sheet, and projecting through the sheet not less than 1 in.; so located as to be at the front of the boiler and exposed to the first pass of the products of combustion.

p In Scotch Marine Type Boilers—in the combustion chamber top, and projecting through the sheet not less than 1 in.

q In Dry Back Scotch Type Boilers—in the rear head, not less than 2 in. above the upper row of tubes, and projecting through the sheet not less than 1 in.

r In Economic Type Boilers—in the rear head, above the upper row of tubes.

s In Cast-Iron Sectional Heating Boilers—in a section over and in direct contact with the products of combustion in the primary combustion chamber.

t In Water-tube Boilers, Worthington Type—in the front side of the steam and water drum, not less than 4 in. above the bottom of the drum, and projecting through the sheet not less than 1 in.

u For other types and new designs, fusible plugs shall be placed at the lowest permissible water level, in the direct path of the products of combustion, as near the primary combustion chamber as possible.

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